POWER

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Why the Ton?

READ the records of the early steam-engine and steam-boiler tests and you will see the results expressed in "bushels" of coal.

John Bourne, writing in 1846, says that "Mr. Watt reckoned his boilers as capable of evaporating 10.08 cubic feet of water with a bushel, or 84 pounds, of Newcastle coal."

And this was the general practice. They had no large scales available for weighing the quantities of coal and water used even in their boilers and engines of limited capacity, so they used the volumes that they could measure.

When weighing apparatus became generally available, they dropped the variable and unscientific bushel.

Then there is the "stone," in which the Britisher gives you his weight today. One stone is 14 pounds, two stones is a "quarter" and four quarters a "hundredweight"—God knows why!

And twenty of these hundredweights make one kind of a ton, 2,240 pounds.

And then somebody had a gleam of sense and established the ton of 2,000 pounds. And it seems to have become popular quickly, and to have superseded the "long ton" for most purposes.

And then there is the metric ton, 1,000 kilograms, or 2,200 pounds.

And if you are dealing outside of some narrow circle where the kind and value of the ton is fixed by law or usage, you have to specify or inquire which ton is meant or valid.

We are gradually dropping the ton altogether, and why should we not?

We used to talk about the "cost per ton" of steam. Now we talk of "cost per thousand pounds."

Many things formerly bought by the ton are now bought by the thousand pounds.

If the ton is to be continued it should be standardized; but why bother? Is it not easier and better to forget it except for those big usages where tonnage means something?

F. R. Low

Redevelopment of Spiers Falls Hydro-Electric Project

By C. VOETSCH
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PLANT on the upper Hudson River, of the Adirondack Power & Light Corporation, which was laid out for ten 5,000-hp. horizontal-shaft units, seven of which had been installed and generated 40-cycle current, is being rebuilt for 60-cycle service with modern type 9,000-hp. vertical-shaft machines to operate under 80-ft. head. The first step in reconstruction has been completed and consists of one 9,000-hp. machine and a high-voltage outdoor substation. Interesting construction problems are involved in changing over the setting intended for the old units to meet the conditions of the new machines.

GREAT activity in water-power development has been going on during the last few years on the upper Hudson River in the neighborhood of Glens Falls, N. Y. The proposal to regulate this river by a large storage dam and reservoir on the Sacandaga River has helped to add interest to developments in this section. In the last three years two new plants have been added, namely, Sherman Island of the International Paper Co. (see *Power*, Nov. 28, 1923) and the plant at Feeder Dam, which is now nearing completion. The two plants are designed to generate 60-cycle current, while the plant at Spiers Falls, of the Adirondack Power & Light Corp., situated directly above these two, generates 40-cycle current. It is contemplated to change this plant and the 40-cycle distribution system for 60-cycle operation in steps as the load may demand.

In the Spiers Falls plant, which is 392 ft. long and divided into 28 bays, 14 ft. wide, there were installed seven 5,000-hp. horizontal-shaft double-runner center-discharge turbines connected to 3,800-kw. generators. Each of the units occupies the space of two bays, so that a machine is installed in one and the other allows space between units. This equipment, although giving satisfactory performance from a reliability standpoint, has not the efficiency and regulation of modern equipment with the exception of units 1, 2 and 3. The present power-house floor is at too low a level and has been flooded a number of times during high-water periods. This has also made a change in design desirable. Furthermore, the proposed superpower system will in all probability operate at 60 cycles and the company's system and power plants will have to change ultimately to this frequency. Therefore it was desirable that 60-cycle units should be added to the present plant in such a manner as to replace the 40-cycle machine at a time when power at 60 cycles is required. As a start toward making this change, one 9,000-hp. vertical-shaft turbine coupled to an 8,500-kva. 60-cycle generator has been installed, with the generator floor raised 10 ft. 6 in. above the present level.

The rebuilding of an old plant is a problem in itself, and conditions must be met as they are. It becomes a case of not what might be desirable to do, but what must be done to get the new equipment installed and at the same time give service with the old. The present power-plant building was originally intended for 10 units of the old type. The remaining space in the building is large enough to install two new vertical machines of the capacity selected, before it will become necessary to start dismantling the 40-cycle section of the plant. One of the new units will occupy the space of two old ones. Only four of the old units are to be replaced, namely, Nos. 4 to 7. Machines Nos. 1 to 3 are to be rewound for 60 cycles, since the wheels on these show a high efficiency. Therefore, when the present plans are completed, this station will contain four 9,000-hp. machines and three 5,000-hp. units, giving a total capacity of 51,000 hp. The present head on the wheels is 80 ft. It may be possible at some future time to raise this to 87 ft., consequently the racks and intake work have been designed accordingly. There is ample room for further extending the plant if it becomes desirable after the present plans are completed.

Constructing a Foundation for the New Unit

As shown in Fig. 1, the foundation for the old power-house consists of two tiers of arches, one supporting the generator and the other the power-house wall on the tailrace side. Otherwise the space under the plant is open, there being no division walls between the units.

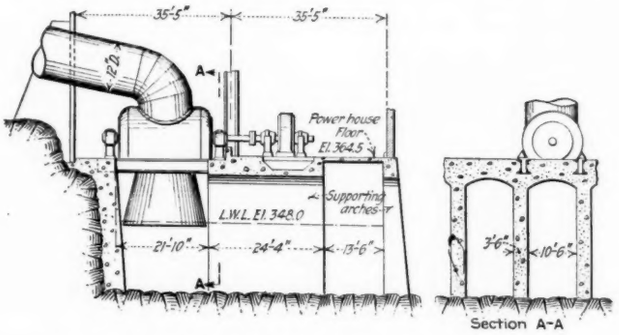


Fig. 1—Section through one of the seven old 5,000-hp. units

The water from the short steel-plate draft tubes can flow in any direction that will form the easiest path to the tailrace.

The chief difficulty in constructing the foundation for the new unit was to create a pit which, after being unwatered, was to receive the draft tube. To provide room for the installation of the second new unit it was necessary that one side of the foundation of the first machine be under the generator of old unit No. 7. Since there were no division walls under the plant, it meant that the site of the foundation for the new unit had to be surrounded on all four sides with a cofferdam before it could be unwatered. A cofferdam 35 ft. high would have to be about 22 ft. wide and would have cost, it was estimated, $80,000 as the mate-

rial was difficult to obtain and an elaborate arrangement of haulage would be required to convey this material across the forebay and through the power house. The tailrace could not be made dry, and the removal of this large quantity of filling material and forms required for a cofferdam would constitute a difficult problem in itself. It was therefore decided that

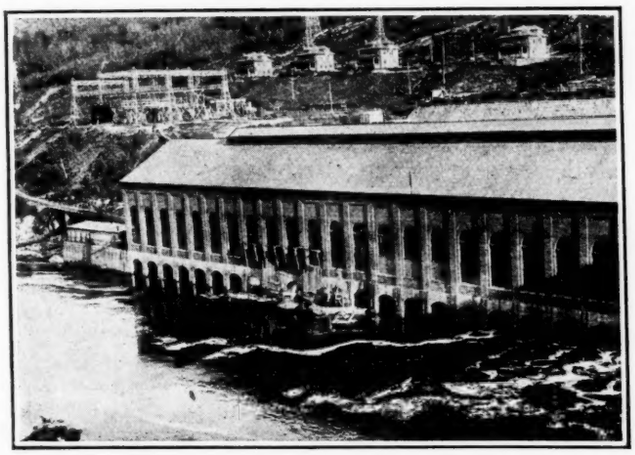


Fig. 2—Power house 392 ft. long and supported on two tiers of arches

the easiest way to create a pit for the draft tube was by placing the outside concrete under water. The arch piers under the plant favored this method of construction and placing the forms at little added expense to that for unwatered conditions.

Creating a Pit for the Draft Tube

Fig. 4 shows the form work and scheme of creating the pit. With this arrangement no more concrete had to be placed than was necessary for the foundation. The forms for the block in the east bay between columns 20 and 21 were placed first, then the block in the west bay between columns 16 and 17. This block extended in three feet between the piers of old generator No. 7 foundation. The form work employed is shown around the block between columns 16 and 17. The frames with a stationary panel and guide rails were sunk first, then the driving planks between the stationary plank and guide rails were driven. The openings left at the bottom by the spacing planks and other small openings at the corners were closed by a diver. As there was very little headroom under the arch of the old generator foundation, between columns 16 and 17 cribwork had to be employed to hold this end of the form. This cribwork was alternately built up a few feet and then sunk by filling with stone. The forms for the rear wall between columns 16 and 21 were placed similar to the side walls. After the wall had been formed, the space under generator No. 7 was filled with concrete, as was also the space under the arch between columns Nos. 20 and 21.

After this work was done, the débris about 8 ft. in depth was removed from the pit where the new draft tube was to be constructed by a single-line clamshell bucket operated from the power-house crane. What could not be cleaned in this way was flushed out into the tailrace by means of a pump delivering about 60 lb. pressure to a 1¼-in. nozzle. This sluicing process proved to be very efficient, considering that the work had to be done under 25 ft. of water, and gave a clean rock surface.

To close the tailrace side of the pit, stop-log guides were first bolted to the downstream side of the columns supporting the arches under the power-house wall. Stop-logs were next put into place, but on account of the rock formation the logs could not be lowered to within about three or four feet of the bottom. This difficulty was overcome by using driving planks outside the stop-logs. Holes for 1-in. anchor bolts were drilled into the rock in the bottom of the pit, and after these bolts were put into place to hold the bottom slab against upward pressure, this slab in the pockets between the cross-walls and stop-logs was poured under water up to about two feet of the draft-tube floor level. The pit was then pumped out and was found to be fairly tight except for some leakage in the corners at the end of the stop-logs, which could be easily closed. The other minor leaks were led to a sump and kept pumped out. The pilasters and arches supporting the power house had to be braced with steel jacks on account of the heavy pressure against the stop-logs and because the arches were badly cracked and had poorly bonded construction joints.

After the pit was unwatered, the next step was to place the final draft-tube floor and the horizontal ring to hold the rear wall against the water pressure. Columns 18 and 19 were backed by an additional 4x5 ft. of reinforced concrete designed as a vertical beam. After this concrete could be subjected to load, the jacks were cut off flush with a cutting torch, with the outside of the columns and the remaining old concrete removed. From here on the construction work was about the same as for any other job of this kind.

Rating of New Turbine

The turbine is rated at 9,000 hp. at 80 ft. head, operates at 163.6 r.p.m. and is guaranteed to have an efficiency of 87.5 per cent at full load, 88.5 per cent at

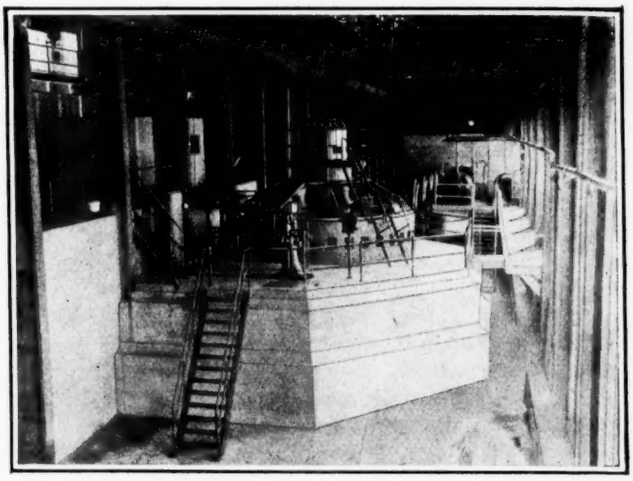


Fig. 3—New generator supported on a watertight reinforced-concrete pedestal 10.5 ft. above power-house floor

75 per cent load and 83.5 per cent at one-half load. It is set in a reinforced-concrete scroll case built into the foundation, and the entire weight of the generator and its foundation is supported by 10 vanes in the speed ring. This weight, including the turbine and draft-tube bell, is transmitted through eight vane-shaped columns extending from the bottom of bell to the floor of the draft tube. The draft tube is of the

Moody, spreading type with the cone extending up to within about seven feet of the runner. In order to obtain room for the scroll-case reinforcement, a cast-iron thimble had to be used at the upper end of the draft tube. Provisions have been made to enter the draft tube through a manhole in this cast-iron section by a tunnel in the foundation that extends from the turbine-room floor.

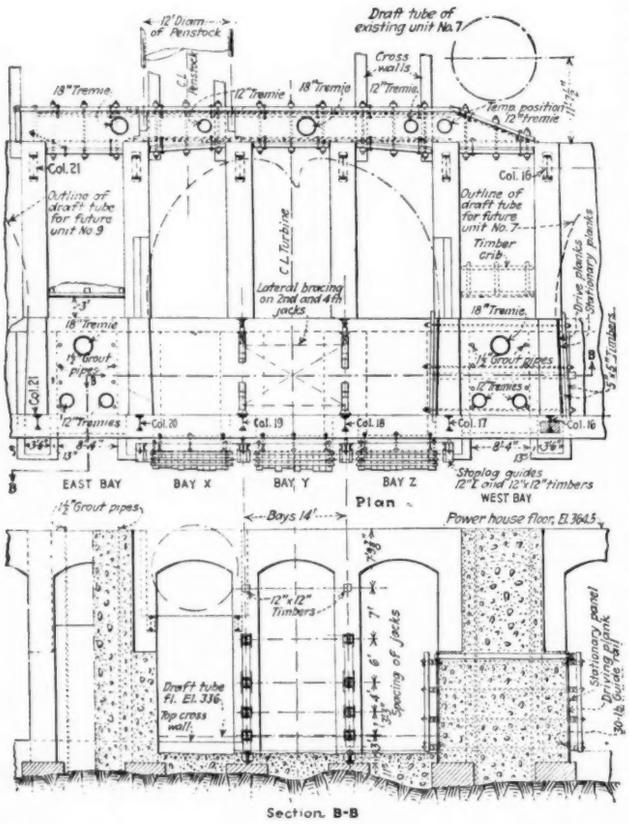


Fig. 4—Concrete poured under water to form pit for draft tube of new unit

Reinforced Concrete Scroll Case Designed for 102-Foot Head

An interesting feature of the turbine is the construction of the scroll case, which is of reinforced concrete designed for 102-ft. water pressure including water hammer when closing the turbine gates in two seconds. The insert, Fig. 6, shows the general arrangement of the reinforcing bars, which are $1\frac{7}{8}$ in. square for the circular and longitudinal reinforcement. The anchoring of the circular reinforcing bars is to four $1\frac{1}{2}$-in. diameter lace rings above and below the speed ring. There was not sufficient room on the speed ring to anchor all the bars, and furthermore the arrangement chosen prevented the concentration of a mass of reinforcing steel at the speed ring. After the construction was completed and the unit put into service, no leaks or even moist spots showed up in the concrete. However, it is probable that 80- to 100-ft. head is about the limit for this type of scroll-case construction, if for no other than economic reasons.

Each of the old units is connected to the forebay by a 12-ft. steel-plate penstock. The plant was originally laid out for 10 units, and the upper ends of the penstocks for all 10 machines were put in the concrete at the headworks, so that when the new work was started there were 3 penstock stubs available in the headworks. As one new unit requires about the space of two old ones, it was therefore possible to use two of the existing intakes in the headworks to supply the new unit. Doing this gave liberal rack area and small entrance losses to the penstocks. The rack sections in each intake are 16 ft. wide with about 33 ft. submerged. Each existing penstock stub had a 30-in. air vent, one of which was enlarged and provided with a ladder to allow access to the upper portion of the penstock and to the rear of the headgates. At the intake the penstocks expand to 13 ft. and are closed by structural-steel gates that travel on rollers, which, when the gate is almost closed, drop into notches, thus bringing the sealing strips together. A motor-operated hoist is used to open and close the gates and can be controlled from the gatehouse or the switchboard room.

Two Old Penstocks Made Into One

The penstock stubs in the forebay were united into a single 12-ft. pipe line by means of 12x12x12-ft. "Y." This "Y" is anchored by a concrete block, Fig. 7. By means of circular reinforcing bars, a 5-in. staybolt and a diaphragm the stresses in the oval section are taken care of properly. From the "Y" the water is led to the turbine through a 12-ft. penstock about 100 ft. long, which is reduced to 11 ft. at the scroll case. All penstock plates are of $\frac{1}{2}$-in. material. After entering the power house, the penstock has to bridge the rear tailrace with a span of 16 ft. Although the penstock can be considered self-supporting over this span, it was found advisable to build a bridge of two 24-in. I-beams, which facilitated the erection considerable and, furthermore, the material was on hand.

A governor of the Pelton type driven by a belt controls the speed of the unit, using oil under a pressure of 150 lb. The governor was tested for a speed change dropping of 3,000 kw., 4,500 kw. and 6,000 kw. load by means of a water rheostat. The latter load, owing to the reduced head on the unit, constituted full-gate con-

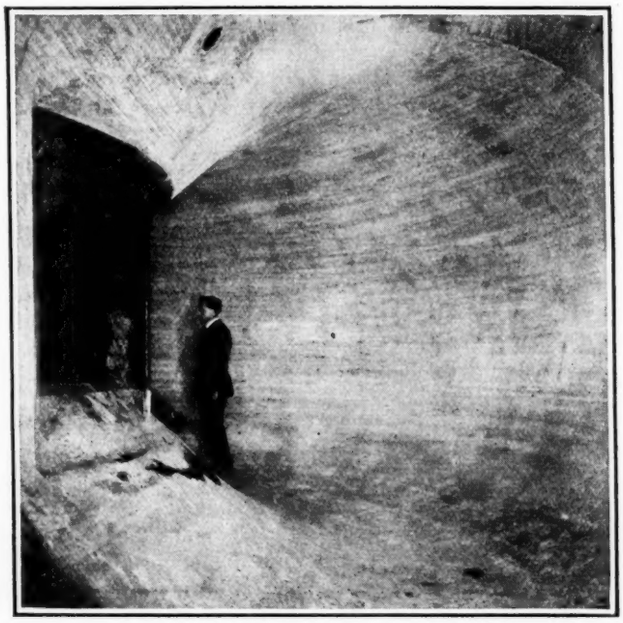


Fig. 5—Inside of reinforced-concrete scroll case designed to stand a total static head of 102 feet

ditions at normal head. The water rheostat was somewhat small and considerable arcing over was experienced. The turbine was on hand control, then shifted over to governor control and the load immediately dropped without giving the governor time to stabilize itself. Although the tests were made under unfavorable conditions, it showed satisfactory results as indicated in Fig. 8. At a later date the inherent speed change was adjusted to about 3 per cent. During the tests the writer made observations in the vent at the intake, but found no surging with the exception of one which was about two feet and settled back to normal.

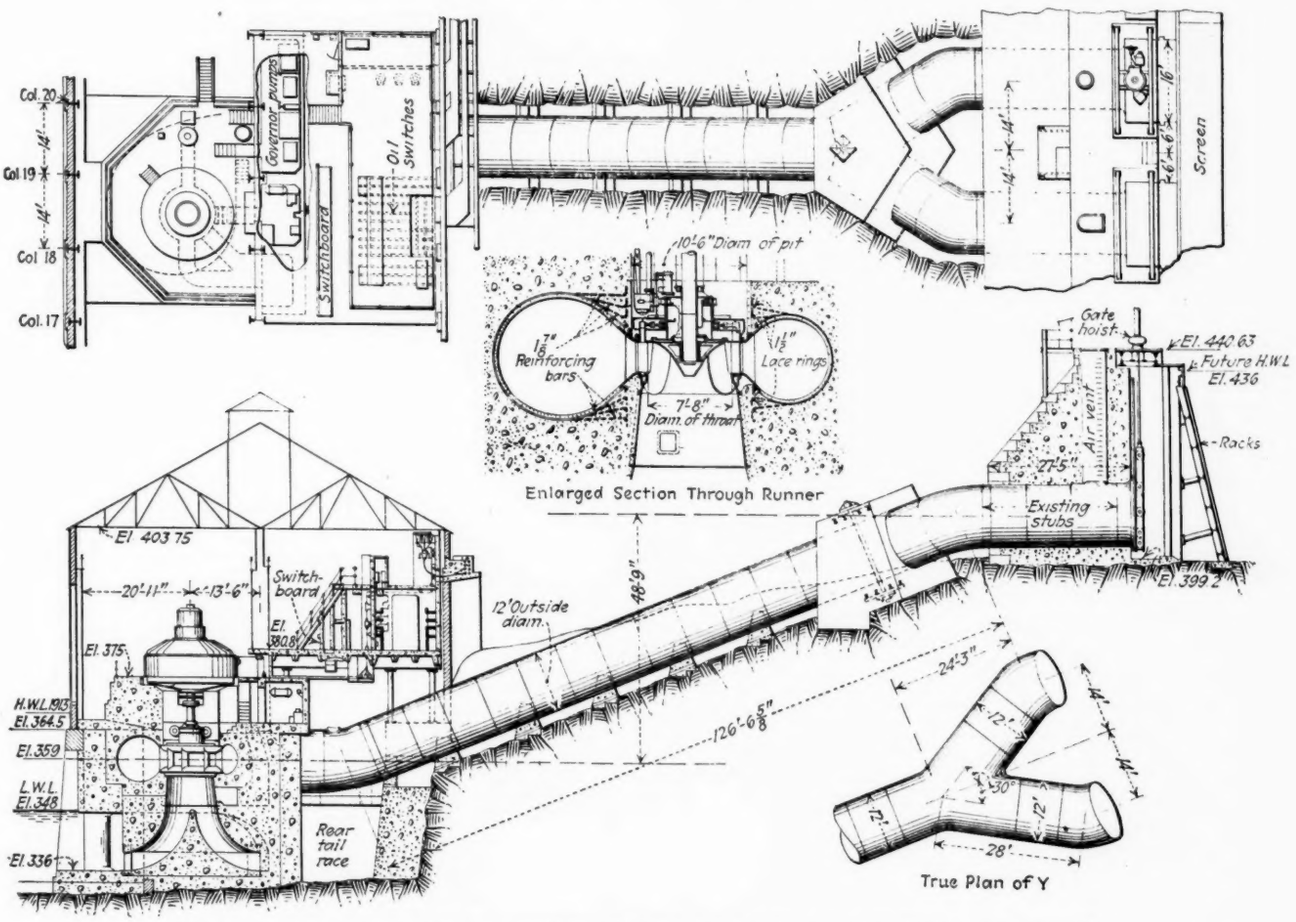


Fig. 6—Two of existing penstock stubs joined into one penstock for new unit

The generator is a three-phase 60-cycle machine having a continuous rating of 8,500 kva. or 6,800 kw. at 80 per cent power factor and 6,600 volts. All six leads of the three phases are brought out and have current transformers connected in them for differential-relay protection. The phases are connected in star with the neutral grounded through a solenoid-operated oil switch.

A spring-type water-cooled thrust bearing is mounted on top of the generator and supports the entire weight of the revolving element including the downward thrust of the water. Excitation is furnished by a direct connected 100-kw. 125-volt exciter.

The regular oil pump for the generator lubrication system is gear driven from the mainshaft. A filter tank and spare direct-current motor-driven oil pump are in the basement. A direct-current motor-driven air compressor and tank designed for 100-lb. pressure for operating the generator air brakes are provided. A connection in the air line permits the application of a portable hand-operated oil-pressure pump when using the brakes as jacks, by which the entire rotating element can be raised high enough to relieve the thrust bearings of its load and free the thrust collar for removal.

In the present 60-cycle development the generator connects to a single bus structure, whence the power is conducted at 6,600 volts to an outdoor substation and stepped up to 66,000 volts for transmission. In the power house the switch cells and bus compartments are installed for the second new unit and provisions have been made for a double-bus arrangement.

For grounding the apparatus and insulator bases, a 1x$\frac{1}{4}$-in. copper strap is used in the entire switch-cell structure and was placed into the forms before pouring the concrete; thus the copper straps are flush with the concrete surface, presenting a neat appearance.

Station service is taken from the main generator bus through an oil circuit breaker from which leads branch to a service transformer bank of three 25-kva. 6,600- to 115- and 230-volt transformers and to a bank of three 50-kva. 6,600- to 230-volt transformers, the latter bank being located at the intake for the operation of the head gates. The former bank supplies the motor of the battery-charging set and the alternating-current motor-driven governor pump with 230-volt current and the lighting system at 115 volts. The lighting system can also be switched to the 40-cycle system of the old plant in case of emergency.

A 60-cell battery supplies the current for the solenoid as well as for the motor-operated oil circuit breakers. A connection to the exciter bus of the old plant supplies the current for the direct-current auxiliaries when first starting up the new unit.

The control and power cables for the outdoor substation are carried from the power house in fiber ducts

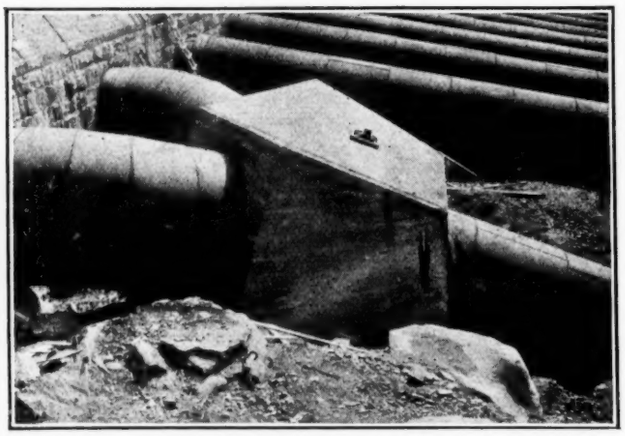


Fig. 7—Penstock "Y" anchored with reinforced concrete block

embedded in concrete and are supported by a steel structure of beams and A-frames, a distance of about 500 ft. Three 3-conductor 500,000-circ.mil varnished-cambric 8,500-volt cables are used per unit between the generator bus and the outdoor station and are protected by differential relays. The structure for the present substation consists of six bays; two for incoming lines from the Sherman Island plant, one transformer bay, one bus tie bay and two outgoing bays to Schenectady. Additional bays can be erected at either end of the station according to future requirements. The station is protected at the incoming as well as the outgoing lines by lightning arresters. For the present one bank of three 6,000-kva. oil-insulated self-cooled transformers is installed for the voltages of 66,000 and 114,000 to 6,600 and 13,200 and is feeding through an oil circuit breaker to a double bus system. The transformers are now connected in delta for 6,600 to 66,000 volts; later, a star connection on the high-voltage side will be used to obtain 114,000 volts.

The electrical installation from the switchboard floor to the outdoor switching and transformer station was designed at the company's office under the direction of C. A. Bacon, electrical engineer. The drawings for the intake, penstock, turbine and generator foundation were prepared at the plant by the writer, who was also

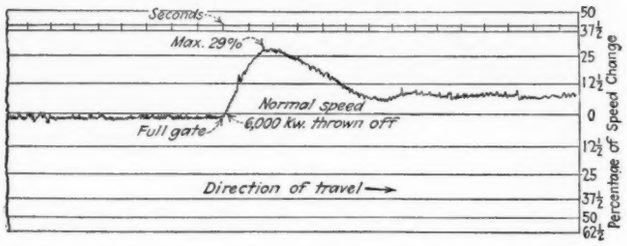


Fig. 8—Speed-change curve when 6,000-kw. load was thrown off the generator

in charge of the entire construction work. L. D. Hoff, civil engineer, and Mr. Himler, mechanical engineer, assisted in detailing the drawings and in laying out of the work.

TABLE OF IMPORTANT DATA OF NEW UNIT IN SPIERS FALLS PLANT OF THE ADIRONDACK POWER COMPANY

Location of general plant	Spiers Falls, N. Y.
Extension made in 1923 and 1924	9,000 hp.
Number of units	1
Ultimate number of units	7, four of 9,000 hp. and three old units of 5,000 hp. each
Full-load Water Velocities Feet per Second:	
Through racks	1.2
Through penstock gates	5
In penstock	10
Axial component from turbine	18.5
In discharge from draft tube	3.2

RACKS, HEADGATES AND PENSTOCK

Racks made in two sections 16x33 ft.
Racks incline with a batter of 1 to 4
Headgates at penstock mouth, roller type made of steel. Gates are motor operated and can be controlled from switchboard.

Size	13.5x14 ft.
Number per turbine	2
Manufacturer of racks and gates and gate-operating mechanism	Treadwell Co.
Electric control for headgates	General Electric Co.

Penstock, 12-ft. diameter ½-in. steel plate.

TURBINES, GOVERNORS AND DRAFT TUBES

Manufacturer of turbine	Wm. Cramp & Sons Ship & Eng. Bldg. Co.
Type	Francis
Number installed, present	1
Number installed ultimately	4
Size	9,000 hp.
Guaranteed efficiency per cent at full load	87.5; three-quarter load 88.5; and half load 83.5
Flow, cu.ft. per second at full load	1,125
Head	80 ft.
Specific speed at normal head of 80 ft.	64.6
Horsepower under 1 ft. head	12.58
Speed, r.p.m.	163.6

Guide bearing, ligum-vitae, water lubricated.
Scroll case, reinforced concrete with cast-iron stay vanes in speed ring.
Draft tube:

Type	Moody spreading
Bottom of draft tube to center of distributor	23 ft.
Diameter at runner discharge	8 ft. 9 in.
Maximum width of chamber at bottom	39 ft. 6 in.
Width of bell at bottom	26 ft. 6 in.
Bottom of bell from bottom of chambers	3 ft. 6 in.
Length of bell from runner	16 ft.
Height of cone	12 ft. 3 in.
Size of discharge to tailrace, 3 openings	11 ft. 6 in. high by 10 ft. 6 in. wide

Governor:

Manufacturer	Pelton Water Wheel Co.
Location	Generator floor

Governors pumps, number 2; motor driven, one by a direct-current motor and one by an alternating-current motor.
Governor fluid, oil under 150 lb. pressure.
Manufacturer of governor pumps: Wm. Cramp & Sons.

Guaranteed Performance of Governor

Load Change, Hp.	Time of Closing Gates, Seconds	Per Cent Speed Change Load Off to 0	Per Cent Speed Change Load On From 0
900	1.204	1.46	1.57
2,259	1.43	4.93	5.12
4,500	1.805	14.05	15.5
9,000	2.5	34.6	54.2

MAIN GENERATOR

Manufacturer	General Electric Co.

Size; 8,500 kva. at 80 per cent power factor, 6,600 volts, 3 phase, 60 cycles.
Guaranteed efficiency, per cent at 80 power factor; full load, 96; ¾ load, 95.5; ½ load, 94.5

Number of poles	44; speed 163.6 r.p.m.
Exciter, direct connected, capacity	100 kw. 125 volts.

Thrust bearing, G. E. spring type, mounted on top of generator.

MAIN TRANSFORMERS

Manufacturer	General Electric Co.
Number installed	3 in one bank

Type, outdoor, self-cooled, oil insulated; voltage 6,600 to 66,000/114,000

MISCELLANEOUS

Storage battery	Electric Storage Battery Co.
Storage-battery charging set	General Electric Co.
Oil Pumps	Brown & Sharpe Mfg. Co.
Lubricating oil filters	S. F. Bowser Co., Inc.
Transformer oil filters	General Electric Co.
Service transformers	General Electric Co.
Switchboards	General Electric Co.
Traveling crane	Northern Engineering Works
Air compressor	General Electric Co.
Twin strainer on turbine guide bearings and cooling coils in thrust bearing	The Elliott Co

Brush Angle and Direction of Commutator Relation

ANSWERS the question, whether a commutator should rotate against or away from the sharp angle of the brush face and shows that a specific direction of rotation is preferable for almost every individual case.

By W. C. KALB
General Manager, Corliss Carbon Company, Bradford, Pa.

SATISFACTORY brush operation is dependent upon several factors directly associated with the manner in which the brush is held against the commutator. In the first place, firm and uninterrupted contact of the entire brush face must be maintained. This necessitates freedom of movement in a radial direction, using the term radial in a broad sense, in order that minor irregularities of commutator surface may be easily followed. Secondly, the relation of the various forces, both active and restraining, applied to the brush must be such that no tendency can arise to change the angle at the brush face. The desirability of keeping friction losses as low as possible constitutes a third factor to be considered, and in some instances the character of the brush material itself places certain restrictions on the position in which it should be held for best results.

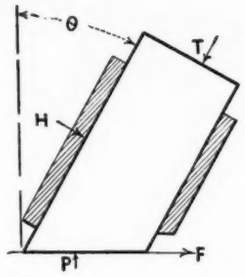


Fig. 1—Brush in leading position

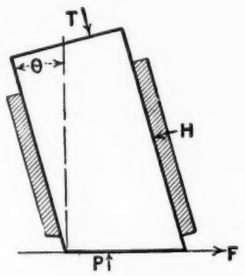


Fig. 2—Brush in trailing position

Forces Exerted on Brushes

In Figs. 1 and 2 are indicated diagrammatically two brushes and the forces exerted on them, Fig. 1 indicating a brush operating in the leading position—that is, with commutator revolving against the sharp angle of the face—and Fig. 2 a brush held in the opposite or trailing position. The forces exerted on the brush are as follows:

- T is the spring tension in a direction parallel to the brush length.
- H is the restraining pressure of the holder against the side of the brush.
- P is the resultant pressure normal to the commutator at the brush face.
- F is the frictional pull at the brush face tangent to the commutator surface.

The latter figure is, of course, the product of P and the coefficient of friction f. By simple equations of components along two co-ordinates, some very interesting relations are disclosed between these forces, the direction of rotation, the angle the brush is inclined from the radial and the coefficient of friction. Without presenting in this discussion the mathematics involved, these relations will be illustrated by means of curves drawn from the formulas so derived.

Fig. 3 shows the value of the ratio $\frac{H}{T}$ at various angles of inclination from the radial with an assumed coefficient of friction of 0.20, the leading side of the holder being considered as presenting a positive pressure against the brush in leading operation and the trailing side in trailing operation. It will be noted that in leading operation there is a tendency for a brush of the assumed coefficient of friction to pull away from the holder at any angle less than 12 deg., but that at greater inclination the reaction against the leading side of the holder exceeds the frictional pull away from it. With trailing operation the value of H is never negative and is always much higher than at the corresponding leading angle. Furthermore, at large trailing angles the wedging effect between commutator and holder becomes pronounced, as is indicated by the tendency of the curve to rise more rapidly with increase of inclination.

Fig. 4 illustrates the variation of the ratio $\frac{H}{T}$ with different coefficients of friction at assumed angles of

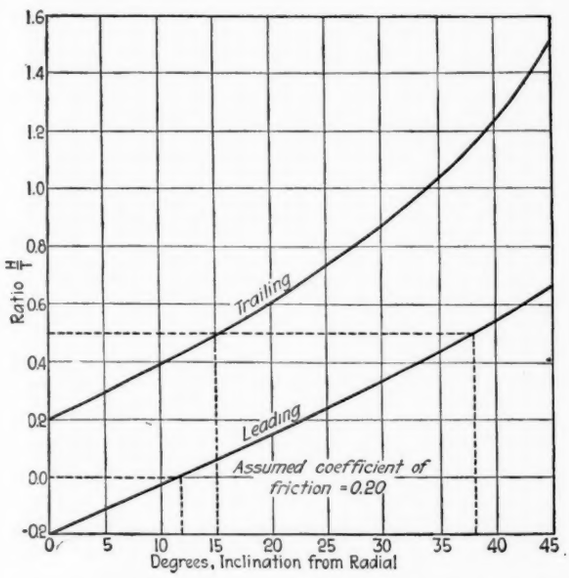


Fig. 3—How brush pressure against the side of the holder changes with a change in brush angle or the commutator

30 deg. leading and 15 deg. trailing. These curves show that at 30 deg. leading angle, firm contact against the side of the holder is maintained as long as the coefficient of friction does not exceed 0.57, and that in the trailing position there is marked increase in pressure against the holder with increase in coefficient of friction.

It has been pointed out that freedom of radial move-

ment is essential to good operation. Naturally, this freedom is not present with heavy pressure against the side of the holder. It is probably fair to assume that this pressure should not be greater than half the tension applied to the brush by the tension spring. From the curves 3 it may be seen that this value of $\frac{H}{T}$ is reached at about 15 deg. inclination in the trailing

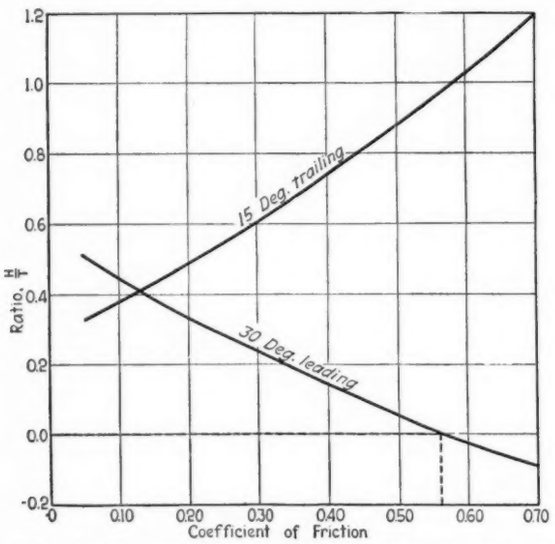


Fig. 4—How brush pressure against the side of the holder changes with a change in the brush coefficient of friction

position and about 38 deg. in the leading position, with a brush having a coefficient of friction of 0.20. With lower coefficient of friction the inclination in the trailing direction may be somewhat greater, but the relationship being reversed in leading operation, it is not advisable to exceed 40 deg. inclination. In recognized practice the trailing angle seldom exceeds 20 deg., while 15 deg. is more common and 37.5 deg. is the largest leading angle in general use with 30 deg. a representative average.

Brush Should Have No Tendency To Tip

The second factor of importance pointed out earlier in this discussion was that there should be no tendency to tip the brush and thus change its face angle. Since the force H never has a negative value in trailing operation, such a tendency can exist only if the lower edge of the holder is too far from the brush face or a disturbing thrust is applied to the spring end of the brush by the tension finger. Sometimes the spring end of the brush is given a slight bevel to prevent the latter contingency. The objection to this practice is that it increases the total pressure against the side of the brush-holder and tends to make the brush sluggish in its radial movement. If the brush length is not more than four or five times its thickness and the tension finger so designed that it has no component of thrust away from the trailing side of the holder, there is little chance for trouble from this source with a machine rotating in only one direction, and top bevel may well be omitted.

In the leading position the angle of the brush on the commutator will not change so long as the force H retains a positive value. However, in determining the angle at which holders shall be set, a factor of safety must be allowed for possible poor commutator surface or high-friction brushes. Experience indicates that chattering of the brushes is likely to be experienced at a 20-deg. leading angle except with perfect commutator surface and brushes of very low friction. Probably 25 deg. is the smallest leading angle that can be safely recommended, and the established practice of 30 deg. to 37.5 deg. is to be preferred. Inasmuch as the value of H is lower in leading operation than in trailing, a bevel at the spring end of the brush may be used without seriously reducing the freedom of radial movement, and this is the almost universal practice on brushes operated in the leading position.

Operate Either Leading or Trailing

From the foregoing it is apparent that brushes may be operated in a satisfactory manner in either leading or trailing position provided the angle of operation is adapted to the direction of rotation and the holder design and adjustment are correct. However, it is generally considered best practice to operate certain soft graphite brushes only in the leading position. This is on account of the susceptibility of such brushes to side wear and the smaller side thrust against the holder in leading operation.

When friction losses become a determining factor, an argument may also be raised in favor of the leading position. Fig. 5, showing the variations of the ratio $\frac{P}{T}$ with coefficient of friction, discloses that the pressure at the contact face increases with coefficient of friction in the trailing position, but decreases with coefficient of friction when the brush is operated leading. For example, when the coefficient of friction is 0.25 with the brush in the leading position, the brush contact pressure is equal to the spring tension in a direction parallel to the brush; with the brush in the trailing position the brush contact pressure is about

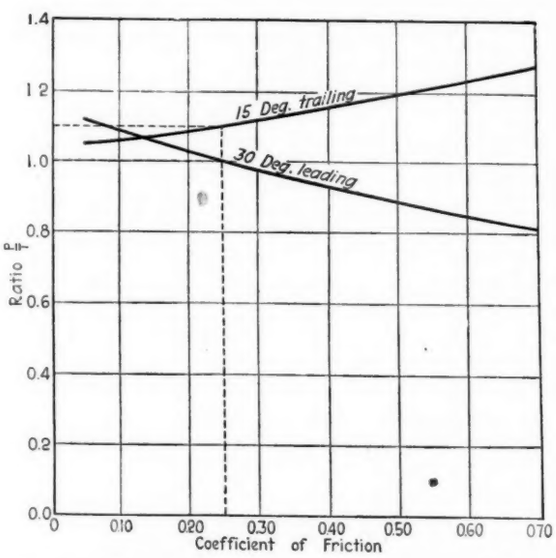


Fig. 5—How brush pressure on the commutator changes with a change in brush coefficient of friction

0.1 greater than the spring tension parallel with the brush.

It is well known that not only does the total brush friction increase with pressure, but the coefficient of

friction itself increases. Consequently, if any condition such as deterioration of commutator surface occurs to increase the coefficient of friction of a brush in the trailing position, the pressure normal to the brush face will also increase, with resultant further increase in coefficient of friction and the multiplication of frictional losses. On the other hand, in the leading position the initial increase in coefficient of friction will be to some extent offset by the reduction of pressure at the brush face, and the total increase of friction loss due to the changed condition kept at a minimum.

From the foregoing discussion, it may be said that if the brush-holder is designed primarily for rotation of commutator away from the sharp edge of the brush face and can be adjusted to an inclination of approximately 15 deg. from the radial but not at an angle greater than 25 deg., trailing operation is preferable to leading; but if the holder design is suitable and permits inclination of 30 deg. to 37.5 deg. from the radial, the rotation of the commutator against the sharp angle of the brush face is generally preferable.

Some Operating Kinks for the Refrigerating Engineer*

By John W. Neff

Each part of the refrigerating plant should be put in perfect working order during the winter so that the danger of a failure during the coming season will be reduced to a minimum. Where local conditions do not make it advisable to do otherwise, the work should be taken up in the following order: Compressors, condensers, ammonia, piping, valves and pipe covering, direct expansion and brine piping, ice-freezing tanks and equipment, raw-water equipment, water-cooling equipment and electrical equipment.

Since the compressor is probably the most important part of the compression system, it is quite obvious that each part subjected to wear should be examined and put in first-class shape. Suction and discharge valves should be taken apart, examined and ground to a perfect seat; tension on spring should be adjusted so as to get quiet action, without wiredrawing. Some engineers recommend heat treating the valves every two years to eliminate the danger of crystallization. If valve cages do not fit perfectly in the cylinder head, they should be ground to a perfect fit.

The piston-rod packing should be removed; if metallic, care should be taken not to damage the rings or get them interchanged. Remove the piston and examine the cylinder walls; if they have a high polish and no grooves in them, there is no need for the cylinder to be rebored.

If shoulders have formed at the stroke ends, they should be ground or scraped down even with the cylinder surface. If rings are quite loose in the grooves, they should be removed. If they make good contact all around when fitted at the point of largest cylinder diameter, have plenty of lap at the joint and are not less than ¼-inch thick at the point of smallest thickness, they need not be renewed.

Should the piston rod be scored or badly worn, it should be trued up in a lathe and a larger packing used. It may be necessary to bush the stuffing box and gland, depending upon the amount turned off rod. Make sure that there is plenty of room for adjustment of wear for the coming season. See that all oil grooves are open and amply deep.

When the piston is out, it is well to line up the mainshaft with the center line of cylinder and the crosshead adjusted to bring its center in line with the cylinder center line. Examine disk and crankpin to see if there are any signs of their being loose. If main stop and bypass valves have soft seats, be sure to examine them; if badly worn, have them renewed. If an automatic oiling system is used on the compressor, be sure to thoroughly clean the entire system.

Clean the condenser pipe of all scale, either by hand or mechanical means, followed by a test with ammonia at 250 lb. pressure. Paint all surfaces not constantly flooded with water with red lead and boiled oil or any good rust-resisting paint. Paint the center of the coils with oil taken from the separator or filter, as a coat of paint will reduce the transfer of heat. Where valve stem packing has become hard, it should be renewed.

Should there be any leaks in the cooling coils, pump out and repair them. Examine the hangers that support the coils, to make sure there is no danger of their giving away when the coils are heavily frosted. While in the storage, it might be well to examine the roof trusses where they join the brick wall, as failures have occurred at that point.

Scraping the Tubes

In case the brine is cooled with a shell-and-tube cooler, the tubes should be scraped clean by means of a flue scraper or wire flue brush or both, as all scale must be removed from the inside of the tubes. After scraping, wash with water under about 75 lb. pressure to remove any loose scale or sediment that may be left in the tubes. Examine the pipe and fitting connections and test with ammonia under 75 lb. pressure. A removable screen should be placed in front of the cooler to prevent small pieces of wood from entering the tubes.

Put the mechanical equipment of the water-softening system in shape by cleaning and making any needed repairs. Remove the sand from the filters, wash or renew it; remove strainers and see that all openings are free of sand or other foreign matter. While cleaning the sedimentation tank, examine the downtake to see that the chemically treated water is not short-circuiting due to openings in the downtake.

Make a thorough examination of the cooling tower for rotten or decaying timbers. In the spring of the year, after freezing weather has passed, the tower should again be inspected to see that falling ice has not broken holes in one or more of the decks. If a spray system is used, see that all nozzles have full opening and are not clogged with small pieces of wood or gravel.

Inspect all compensators, controllers, oil switches and automatic starting devices. All contacts should be kept clean and, if badly burned, should be renewed. The oil in oil switches should be renewed each season. Keep the interior as well as the exterior of the motors clean by blowing out with low-pressure air. Bearing linings should be inspected and, if badly worn, renewed, as there is danger of the rotor rubbing on the stator.

*Extract of paper read before the annual meeting of the National Association of Practical Refrigerating Engineers at New Orleans, Nov. 13.

Attention has been called to a typographical error in the title of L. A. Quayle's article on page 844 of the November 25 issue. The title should read "Fairmount Pumping Station and Heating Plant."

Determining the Leaving Loss and Limiting Vacuum of a Steam Turbine—I

By B. C. SPRAGUE*

Economical turbine operation depends to a large extent on supplying the degree of vacuum conducive to best over-all plant efficiency. In Part II, which will appear in an early issue, a method of determining the leaving loss for a turbine is taken up. Discussion of the points brought out is invited.

IN ORDER to obtain the best possible economy in a power station, it is necessary to find the most economical methods of operation by making a thorough study both of the equipment and of the limiting operating factors. Not only do turbines differ greatly in their ability to utilize high vacuums economically, but sources of cooling water differ in their tendencies to maintain vacuum, due to the fouling of condenser tubes. The temperature of the circulating water is another factor that must be taken into consideration, as this changes from day to day throughout the year. It is necessary, therefore, to vary the volume of cooling water and the frequency of condenser cleaning if the most economical operation is to be secured. In determining the best operating methods, it is essential to know what change in turbine economy will result from a unit change in vacuum at the various operating back pressures.

Although a slight rise in vacuum will, generally

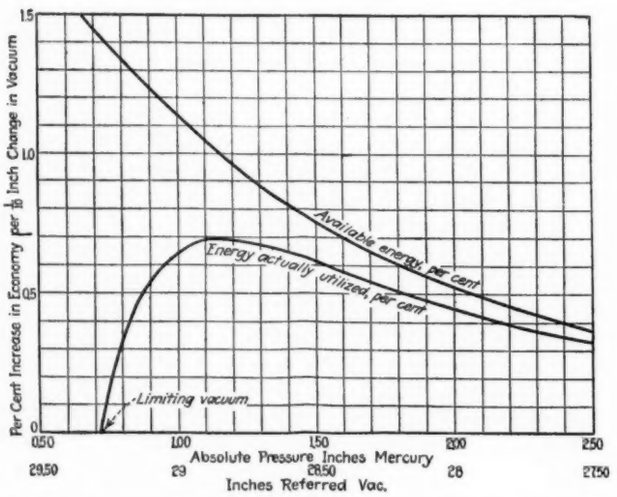


Fig. 1—Decreasing the exhaust back pressure improves economy only up to the limiting vacuum

speaking, lower the turbine water rate, it is also true that the corresponding rate of improvement in steam consumption increases only over a certain range of vacuum. It reaches a maximum and then decreases sharply as the vacuum is further raised. Fig. 1 shows the energy actually utilized as well as that available in a certain turbine operating at rated load. It will be seen from the curves that the rate of change in steam consumption with regard to vacuum first increases as the vacuum is raised and reaches a maximum at 28.85 in., then declines to zero at 29.28, this point being called the limiting vacuum. This limiting value, as

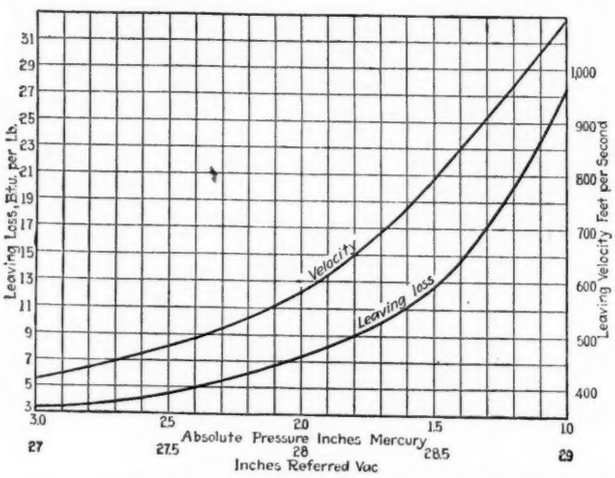


Fig. 2—How leaving loss and velocity varies with vacuum in a 20,000-kw. turbine at full load

well as other vacuum corrections, will depend on the type of turbine and the load carried.

It is the purpose to show how, by the use of curves, the limiting vacuum and the vacuum correction can be found for the various operating loads on any turbine.

A better vacuum at the exhaust means that more heat drop or energy is made available by the steam expanding in the turbine. The upper curve, Fig. 1, shows the per cent increase in available heat drop by adiabatic expansion per tenth inch increase in vacuum for steam with an initial pressure of 300 lb. gage and 200 deg. F. superheat. In an ideal turbine, the per cent change in steam consumption with better vacuum would be as shown in this curve. Owing to the effect of the higher vacuum on the losses in actual practice, the resultant change in steam consumption for a given case is as shown in the lower curve. It will be seen that the difference between these curves becomes greater as the vacuum increases as at the left of the figure. This is due to the greater turbine losses.

Raising the vacuum at the rated most economical load, besides increasing the available heat drop, tends to affect the turbine economy as follows:

1. Increases the amount of work done by the wet steam, tending to lower the turbine efficiency, meaning the Rankine cycle ratio, or ratio of work done to that made available by expansion of the steam. Friction losses are relatively large owing to the increased water in steam in the region of last expansion.

2. Increases the heat returned to the steam through losses and hence the reheat factor, tending in some respects to improve, but on the whole to lower, the turbine efficiency.

*Efficiency Department, West Penn Power Co.

3. Results in higher steam velocity in the blades, giving a higher ratio of relative steam speed to blade speed, and thus lowers the turbine efficiency.

4. Makes the volume of the steam greater on leaving the last row of blades with a consequent increase in the leaving loss. The effect of the first and second items are small, and since the first tends to increase and the second to lower the turbine efficiency, they may be neglected.

It is well known that to set any mass in motion, an amount of energy is required proportional to the square of the speed or velocity. This energy remains "stored up" in the moving mass until turned into work upon reducing the velocity. In the case of a jet of steam, the energy required to give motion to each pound is

$$E = \frac{V^2}{2g}$$

where E is the energy in foot-pounds, V is the velocity in feet per second, and g is the acceleration due to gravity of 32.2 ft.-sec. The heat equivalent of the energy per pound of steam is

$$H = \left(\frac{V}{223.7}\right)^2$$

when H is the energy in British thermal units.

The kinetic energy stored in the steam leaving the last row of blades of a turbine is practically all converted into heat owing to eddying in the exhaust space, and thus lost as far as doing any work in the turbine is concerned. This loss has been termed the "leaving loss." The velocity of the steam leaving the last row of blades of a given turbine, and hence the leaving loss, increases as the vacuum at the last row of blades is raised, and as the weight of steam being exhausted is increased. This loss is also larger for any given capacity, as the area of the last row of blades is reduced. Fig. 2 shows the leaving velocity and leaving loss of a 20,000-kw. turbine operating at full load, as occurs at various vacuums.

The blade angle of the last row of moving vanes in a turbine is designed so as to cause the path of the leaving steam to be in an axial direction, or in other words, the vanes discharge steam parallel with the shaft, for a certain load and exhaust pressure. At other loads and exhaust pressures, however, the direction of the leaving steam may be at a slight angle from the axial. This matter is again taken up in Part II, wherein a method has been developed for calculating vacuum corrections, and limiting vacuum.

Selecting Guys for Steel Chimneys

By W. WALLACE CHRISTIE*

IT IS NECESSARY first to decide upon the highest expected wind velocity, in order to calculate the total load to be borne by the guys, as described in the March 20, 1923, issue. The latter is obtained by dividing the horizontal wind-pressure load by the cosine of the angle made by the guy wire with a horizontal line. A perfectly tight wire without sag is impossible. To allow for the effect of the slack, one authority suggests increasing the calculated load by 15 to 20 per cent. Snow load should not exceed two pounds per foot of wire and is usually negligible; clip connection with thimble also reduces the calculated capacity of wire rope guys by 15 per cent, which, with the foregoing would suggest using 30 per cent increase in the total load to be taken care of.

This should be used in connection with a factor of safety of three. Thus, $1.00 + 0.30 = 1.30$; then $1.30 \times 3 = 3.90$. The maximum actual guy load times 3.9 should be the breaking strength of wire ropes as selected from the tables.

From Fig. 1 we can determine what proportion of the horizontal load will be taken by a single guy if we know the number of guys on the chimney. Fig. 2 shows what proportion of the guy is effective. The actual guy load, or stress, equals the maximum horizontal required pull divided by the factor obtained at the left.

It was a matter of much surprise to the author to find such a variety of opinions existing among wire manufacturers, as to how the size of chimney guys should be determined. For example, one writes: "For above 125 ft., we recommend four sets of three guys each, all guys ½ in. in diameter. This will give ample strength and allows for considerable corrosion." Further is written: "Every boilermaker and stack builder seems to have a rule of his own; for instance there is a steel stack near our factory 4 ft. 6 in. in diameter by 125 ft. high. There are only eight guys, ⅜ in. in diameter, four on a band three-fifths of the height and four on another band four-fifths of the height, leaving no margin of safety whatever, even if good steel wire rope is used." The stack was erected in 1921.

This wire manufacturer suggests the following rule: Maximum stress in guys, due to wind and initial stress $=$ wind pressure $\times D \times H \times \frac{H}{h_1}$ where D is the chimney diameter in feet, H the chimney height in feet, and h_1 the height in feet to upper guy band. Note that no account is taken of the angle of wire ropes to the horizontal in this rule, which should be done as before referred to.

Another wire maker says that when guys and their anchorages are equally spaced about a chimney, the maximum value of the guy factor, for example, with six guys is 2; that is, for any position of the wind or load the guys must have two times the average load if divided equally among the guys on one-half of the chimney. An odd number of guys is always the best to use for economy's sake. If seven is the number, then 2.248 is the guy factor. This company recommends four as the factor of safety.

In connection with Fig. 1 a manufacturer advises that it is best always to use an odd number of guys in guying a piece of apparatus of any size. This is because the maximum and minimum values of the guy factor are very close together for an odd number of guys, whereas with an even number of guys there is a much lower minimum value. For example, a derrick employing 6 guys has a minimum guy factor of 1.732.

The addition of one guy or increasing the guys by one-sixth will raise the value of the guy factor to 2.248, an increase of 30 per cent. In the interest of economy

*Consulting Engineer, Ridgewood, New Jersey; Author "Chimney Design and Theory."

it is always advisable, therefore, to use a large number of guys. It is further essential that the guys be spaced evenly so that the angle between any pair is the same as that between any other pair.

To determine quickly the size of wire used in the manufacture of seven-wire strand, divide the decimal diameter of finished strand by three; for nineteen-wire strand divide by five.

The popular sizes are $\frac{5}{16}$ in. and $\frac{3}{8}$ in. diameter and are what is known as galvanized steel guy strand.

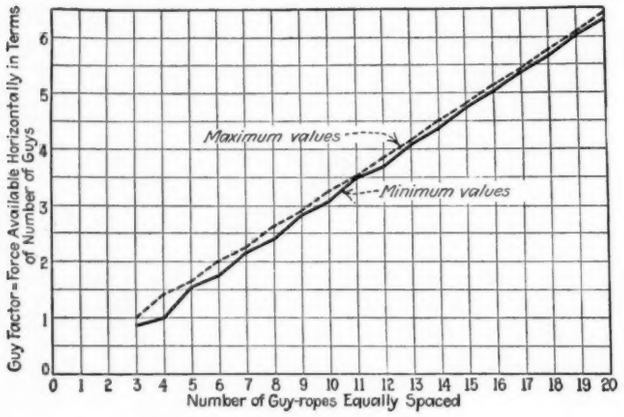

Guy Factor = Force Available Horizontally in Terms of Number of Guys
Maximum values
Minimum values
Number of Guy-ropes Equally Spaced

*Fig. 1—Horizontal force per guy equals total force divided by guy factor at left**

These are evidently meant for small chimneys only, approximate strengths being given in Table I.

Another maker advises that wire of six strands, seven wires each, specified as haulage rope is used for guys and ship rigging. He gives for ultimate strengths of classifications named in Table I as (1) 75,000 to

TABLE I—GALVANIZED STEEL STRAND
Approximate strength and weight in pounds

Diameter	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$
1. Standard steel	2,300	3,800	5,000	6,500	8,500	11,000	14,000
2. Siemens-Martin	3,060	4,860	6,800	9,000	11,000	15,000	19,000
3. High strength	5,100	8,100	11,500	1,500	18,000	21,500	25,000
4. Extra high strength	7,600	12,100	22,500	22,500	27,000	35,000	42,500
Approx. wt. 1,000 ft.	125	210	295	415	510	650	800

100,000 lb. per sq.in.; (2) 150,000 to 200,000 lb. per sq.in.; (3) 180,000 to 220,000 lb. per sq.in., and (4) 200,000 to 260,000 lb. per sq.in. Table II gives comparative strengths of steel wire rope.

Clip connection is approximately 85 per cent efficient, and the lock of a thimble still further reduces efficiency. Either the rope or strand must be galvanized as its

TABLE II—COMPARATIVE STRENGTHS OF STEEL WIRE ROPE IN TONS OF 2,000 LB.

	$\frac{3}{8}$ In.	$\frac{1}{2}$ In.	$\frac{3}{4}$ In.	$\frac{7}{8}$ In.	1 In.
$\frac{6}{19}$ Cast steel	4.6	7.7	13	18.6	24	31
$\frac{6}{19}$ Ex. str. steel	5.25	8.85	14.5	21.0	28	35
$\frac{6}{19}$ Plow steel	5.9	10.0	16.0	23.0	31	38
$\frac{6}{7}$ Cast steel	4.8	8.4	12.5	17.5	23	30
$\frac{6}{7}$ Ex. str. steel	5.3	9.2	14.0	20.2	26	34
$\frac{6}{7}$ Plow steel	5.75	10.0	15.5	23.0	29	38

further or better protection would then mean no loss in strength for the metal. Three should be the minimum factor of safety, and five should be a maximum, with four for an average. Galvanized ships rigging and guy ropes are made six strands of seven or nineteen wires each, with a hemp center. Table III gives approximate strengths of galvanized and cast steel guy ropes, composed of six strands and a hemp center, with seven or nineteen wires to the strand.

*From catalog of the American Steel & Wire Company.

To secure the end of the guy away from the chimney, a "deadman" of wood or steel is usually employed. The fact that all the earth above it must be loosened up by digging a trench or hole, and the possibility of the trench being too shallow, makes this method not altogether a desirable one. It is still used for many kinds of anchorage, especially for the heavier strains.

To meet objections to the rotting of buried timber, and avoid as much as possible the loosening up of earth,

TABLE III—PROPERTIES OF GALVANIZED CAST STEEL GUY ROPES
Composed of six strands and a hemp center, with seven or nineteen wires to the strand

Diameter In.	Approximate Circumference, In.	Approximate Weight per Foot. Lb.	Approximate Strength in Tons of 2,000 Lb.	Circumference of Equal Strength Manila Rope
1	3	1.58	28	9
$\frac{15}{16}$	$2\frac{3}{4}$	1.20	22	$8\frac{1}{4}$
$\frac{7}{8}$	$2\frac{5}{8}$	1.03	19	8
$\frac{13}{16}$	$2\frac{1}{2}$	0.89	16.8	7
$\frac{3}{4}$	2	0.62	11.7	6
$\frac{11}{16}$	$1\frac{3}{4}$	0.50	9	$5\frac{1}{2}$
$\frac{5}{8}$	$1\frac{5}{8}$	0.39	7	$4\frac{3}{4}$
$\frac{9}{16}$	$1\frac{1}{2}$	0.34	6	$4\frac{1}{2}$
$\frac{1}{2}$	$1\frac{1}{4}$	0.30	5	$4\frac{1}{2}$
$\frac{7}{16}$	$1\frac{1}{4}$	0.22	4.2	$3\frac{3}{4}$
$\frac{3}{8}$	1	0.15	3.2	3

galvanized anchors have been devised, which can be screwed into the ground.

Steel or iron plates are, where necessary bolted to the sides or framing of wood, steel or concrete structures. Such a plate should have an eye with a thimble in it to fit the wire rope, in order to get the best efficiency in the carrying capacity of the rope.

Braces or latticed frames are used between steel chimneys when placed close together to take the place of some of the guys that would otherwise be necessary.

From experiments by R. C. Carpenter there is the empirical formula $R = 100\ dh^2$, for the force available in an anchor, R, in lb. is the force necessary to pull up a disk or plate buried in firm earth; d is inches in

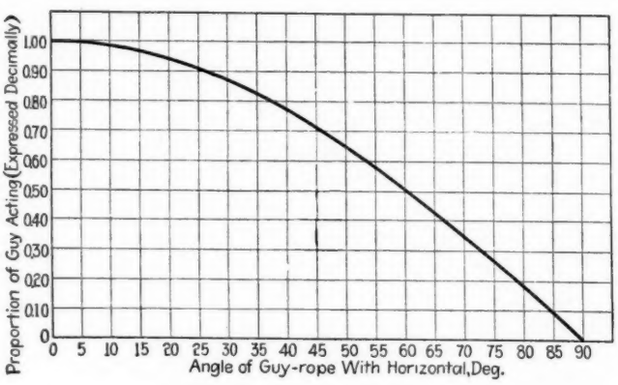


*Fig. 2—Horizontal guy load divided by factor obtained at the left gives full guy stress**

diameter; h is the pull in feet, to be parallel to the axis of the disk or plate.

The foregoing may be either a flat plate buried or a screw-shaped plate screwed into the earth. An 8-in. disk buried 3.33 ft. required a force of 9,000 lb. for its removal. In clay the result R may be increased by 20 per cent. This is useful for guy wire anchoring, by using a factor safety in connection with it.

In the operation of oil-fired boilers precaution should be taken to see that all the burners are consuming approximately the same amount of oil. The operation of the furnace is or can be correctly balanced and completeness of combustion obtained. only when such is the case.

Boiler Feed Pump Explanations and Calculations

By CHARLES L. HUBBARD
Consulting Engineer, Boston, Mass.

Tells what principles underlie successful operation and how to figure capacity, suction lift or pressure, and other things concerning a feed pump. An article, "Selecting Boiler Feed Pumps," will supplement this in a later issue.

DWELLING briefly on fundamental principles, the ability of a pump to lift or raise water from a lower level by suction is due primarily to exhausting the air from the supply pipe, and the forcing of water into the vacuum thus formed, by atmospheric pressure. Under normal conditions at sea level, the latter amounts to 14.7 lb. per sq.in. In practice it is customary first to remove the air from the suction pipe by "priming" the pump, that is, filling it with water under a pressure sufficient to drive out the air. The action, then, is like that shown in Fig. 1, at *A*, where priming water has entered, driving air out of the suction valve in the piston. The priming line appears below the piston. Now, raising the piston creates a vacuum between the bottom of this piston and the surface of the priming water. The downward air pressure forces the supply water from below into this space.

One Foot Water Column Exerts 0.433 Lb. Sq.In.

In Fig. 1 at *B* is a pipe one foot in height and of such diameter that its sectional area is exactly one square inch. If this is filled with water at a temperature of 62 deg. F., it will hold 0.433 lb. In other words, a column of water one square inch in area weighs 0.433 pound per foot in height. Let us now take a pipe of the same size, say forty feet in length and immerse the lower end in a vessel of water that is open to the atmosphere, and exhaust the air from the upper end, as shown at *C*. The atmospheric pressure is assumed to be normal, or 14.7 lb. per sq.in.

With a perfect vacuum it is evident that cold water will rise in the pipe to a height of $\frac{14.7}{0.433} = 33.95$ ft., because the air exerts a pressure of 14.7 lb. upon each square inch of the surface water and will therefore support a column inside the pipe having a weight equal to this pressure. Stated differently, a column of water 1 sq.in. in area and 1 ft. in height weighs 0.433 lb., and so must be $\frac{14.7}{0.433} = 33.95$ ft. high in order to weight 14.7 lb. and balance the atmospheric pressure. This is the principle of pump suction, and theoretically it should be possible to lift water to this height.

TABLE I—TEMPERATURES AT WHICH WATER BOILS AT PRESSURES BELOW ATMOSPHERE

Temperature of Water, Deg. F.	Vapor Pressure, or Boiling Pressure in Lb. per Sq.In. Abs.
40	0.12
62	0.28
80	0.51
100	0.95
150	3.71
200	11.52
212	14.70

In practice this cannot be accomplished for various reasons. It is not possible to maintain a perfect vacuum in the suction pipe owing partly to leakage around the valves and other moving parts of the pump. Water boils and gives off vapor at pressures below atmosphere. We are familiar with this, as steam generation at higher temperatures and pressures, but it also exists at temperatures below the boiling point or atmospheric pressure (212 deg. F.), as shown in Table I.

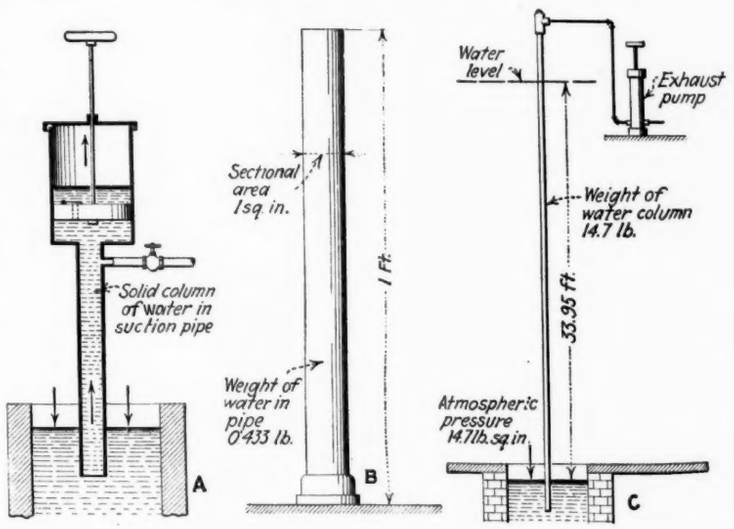


Fig. 1—Why water is raised by suction to a height limited by atmospheric pressure

Raising the piston at *A* creates momentarily a vacant space over the primed water column. An atmospheric pressure of 14.7 lb. per sq.in. forces cold water into this space but can force it only 33.95 ft. high as at *C*, since each foot of water of one square inch cross-section *B* weighs 0.433 lb.

For accurate computations the vapor or steam pressure must be added to the weight of the water column or, what amounts to the same thing, deducted from the atmospheric pressure. If, in Fig. 1 at *C*, the temperature of the water in the pipe is 62 deg. F., the vapor pressure above it will be 0.28 lb. per sq.in., and this will balance or neutralize an equal amount of atmospheric pressure, leaving only 14.7 — 0.28 = 14.42 lb. effective. The latter will support a column only $\frac{14.42}{0.433} = 33.3$ ft. high instead of 33.95.

The friction in the pipe also has an effect. Furthermore, as we ascend from sea level, the atmospheric pressure decreases, so that with a pump installed at an altitude of 1 mile, or 5,280 ft., it becomes only 12.02 lb. per sq.in., and the suction lift due to the atmosphere is reduced to 27.76 ft.

Under practical working conditions, 22 ft. suction lift is usually considered about the limit for a temperature of 60 to 80 deg. at the sea level, and this is constantly reduced as the temperature rises, due partly to the vapor pressure, as previously noted, and also to the air contained in the water. Referring to Fig. 2, the "theoretical curve" at the right shows what suction lift might be expected at sea level with water at

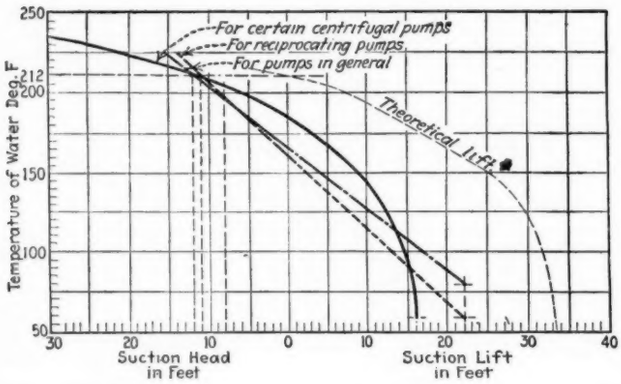


Fig. 2—Theoretical and practical pump-suction conditions as limited by boiling temperatures

On account of the boiling or vapor pressure at temperatures below 212 deg. F, water should be raised by suction as indicated by the theoretical lift, for various temperatures. The presence of air in the water, friction, valve leakage, etc., reduces this in practice to values shown by the remaining lines.

different temperatures, as a result of the vapor or boiling pressure.

In actual practice the results are greatly changed by the effect of the air contained in the water, and the effects of friction in pipes and valves, valve leakage, etc., as shown by the straight lines. One of these is given by Harding and Willard and applies in a general way to all pumps, as no special type is mentioned. This gives a maximum suction lift of 22 ft. at a temperature of 80 deg. F. At 165 deg. and above a suction head or intake pressure is required, increasing to 12 ft. at 212 deg. F. Another is from Marks Handbook and appears under reciprocating pumps. This gives a maximum lift of 22 ft. at 60 deg., crosses the zero point at a temperature of about 160 deg., and requires a head of 11 ft. at 212 deg.

There is plotted from a pump manufacturer's instruction book, a curve which applies to centrifugal pumps ranging from 2 to 12 in. suction. This gives a maximum lift of 16 ft. at a temperature of 60 deg., crosses the zero point at 186 deg., and requires a head of 8 ft. at 212 deg., and 10 ft. at 220 deg., which is the limit given. The straight lines stopped at 212 deg. on the diagrams from which they were taken, but they have been extended to 225 and the curve extended to 30 ft. head.

All three of these graphs show that a pump taking water at a temperature of 212 deg. from an open heater requires a suction head or intake pressure of practically 12 ft. to prevent it from becoming "steambound" or filled with vapor. Also, this head is sufficient for any temperature, because water cannot exist at a temperature above 212 deg., except under a pressure greater than that of the atmosphere, the effect of which is added to the static head.

For example, if 12 ft. suction head is required for water at 212 deg. from an open heater, raising the temperature to 230 will require a gage pressure of approximately 6 lb. per sq.in., which is equivalent to adding $\frac{6}{0.433} = 13.8$ ft. to the head produced by the height of the water column, making a total of $12 + 13.8 = 25.8$ ft., which is considerably in excess of that called for by the straight lines. Therefore the extended curve is preferred for all classes of pumps at temperatures above 220 deg. F. It is evident that 12 ft. head is ample for a boiler feed pump taking its supply from an open heater. If a closed heater was used, supplied with live steam for obtaining a higher temperature, either for boiler feeding or industrial purposes, the water would be forced through it, thus heating on the discharge side, and the suction should be supplied to the pump at the lower temperature before being heated.

While a piston or plunger pump will handle water at any temperature as long as there is sufficient pressure to force it into the cylinders, a centrifugal pump may have difficulty from vaporizing within the casing, if the water is near the boiling point. If this occurs the pump will fail to work, as it depends upon the action of the runner upon a liquid.

A pump raising water from one level to another level 20 ft. above it, as in Fig. 3, is operating under a head of 20 ft., neglecting friction and certain other factors to be mentioned later. The total head against

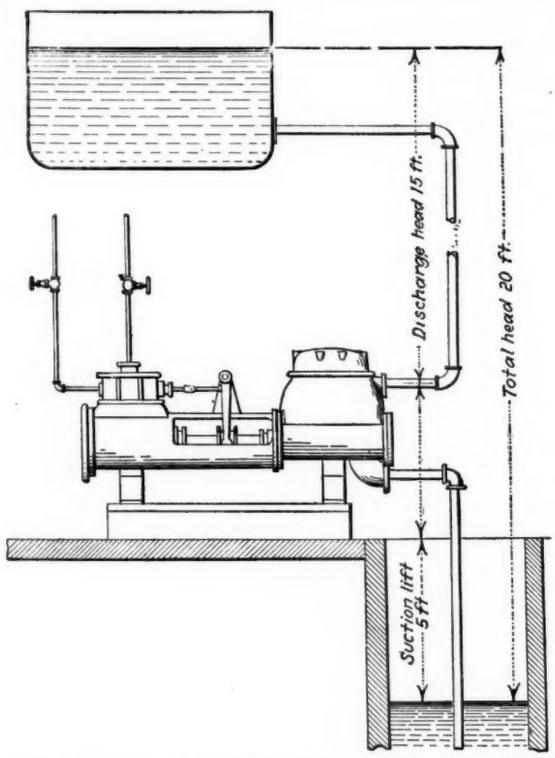


Fig. 3—Difference in suction- and discharge-water levels equals total pressure head

which a pump operates is made up of the pressure or measured head, the velocity head and the friction head. The pressure or measured head is the vertical height from a lower to a higher plane in a pumping system, as from the surface of the reservoir at the base of a cooling tower to the center of the discharge pipe from the circulating pump, at the top of the tower.

When the pump discharges against a pressure due to other causes than elevation, as in feeding a boiler,

the pressure is usually expressed in equivalent feet of head. That is, forcing water into a boiler against 100 lb. gage pressure is equivalent to operating under $\frac{100}{0.433} = 231$ ft. head when referred to a column of water having a temperature of 62 deg. While this is sufficiently accurate for most practical work, it should be borne in mind that the weight of a column of water having a given height varies slightly with the temperature, as shown in Table II.

While this is not of special importance in pumping problems, it makes a decided difference in the operation of certain instruments and regulating devices depending upon the pressure head of water at different temperatures. It is evident from the foregoing definitions that the measured head includes both the suction lift and the delivery lift (20 ft. in Fig. 3). This may be obtained by measurement when elevations only are involved, and by pressure gages when either elevations or pressures, or both, enter into the problem. In this case gages are placed on both the intake and discharge sides of the pump, at the same level if possible.

TABLE II—HOW TEMPERATURE AFFECTS THE WEIGHT OF COLUMN OF WATER OF THE DIMENSIONS AS AT *B*, FIG. 1

Temperature of Water	Weight of Column Having 1 Sq.In. Sectional Area and 1 Ft. in Height
60 deg.	0.433 lb.
100 deg.	0.430 lb.
150 deg.	0.425 lb.
200 deg.	0.417 lb.
300 deg.	0.391 lb.

If the intake gage indicates a pressure below that of the atmosphere (vacuum), it should be added to the reading of the delivery gage, expressing both in the same units. If it indicates a pressure above the atmosphere, it should be subtracted in order to obtain the net pressure head against which the pump is operating. The velocity head or pressure is that required to set the water in motion and maintain the given velocity of flow through the pipes. This is given by the formula $h_v = \frac{v^2}{64.4}$ in which h_v is the velocity head, in feet of water column, and v is the velocity, in feet per second.

As the velocity in boiler feed piping is commonly limited to about 400 ft. per min., or 6.6 ft., per sec., the value of h_v amounts to only $h_v = \frac{6.6 \times 6.6}{64.4} = 0.67$ ft., and may usually be neglected in work of this kind.

Table III gives the friction head for smooth straight wrought-iron pipes 100 ft. in length for velocities within the usual range of boiler feeding, which runs from about 3 to 6 ft. per sec. For other lengths, increase the friction head in proportion, that is, for 200 ft. multiply the values given in this table by 2, and so on.

TABLE III—FRICTION HEAD IN FEET FOR SMOOTH WROUGHT IRON PIPES, 100 FT. IN LENGTH

Size Pipe	1 In.		1¼ In.		1½ In.		2 In.		2½ In.		3 In.	
Gal. per Minute	Vel., Ft. per Sec.	Friction Head, Ft.	Vel., Ft. per Sec.	Friction Head, Ft.	Vel., Ft. per Sec.	Friction Head, Ft.	Vel., Ft. per Sec.	Friction Head, Ft.	Vel., Ft. per Sec.	Friction Head, Ft.	Vel., Ft. per Sec.	Friction Head, Ft.
10	3.7	8.4										
15	6.1	18.9	3.9	4.7								
20			4.2	7.9	3.2	3.7						
25			5.4	11.9	4.6	5.6						
30			6.4	16.9	4.7	7.8	3.1	2.7				
35					5.5	10.3	3.6	3.7				
40					6.3	13.3	4.1	4.7				
45							4.6	5.8	3.0	2.0		
50							5.1	7.1	3.3	2.4		
75									4.9	5.1	3.4	2.1
100									6.5	8.6	4.5	3.5
125											5.7	5.4
150											6.8	7.7

The frictional resistance from the other sources enumerated is commonly expressed in equivalent feet of straight pipe, which is added to the length of run, when using Table III. Table IV gives data of this kind.

Example: A pump is to feed 25 gal. of water per minute into a boiler, against 130 lb. gage pressure. The total height from the surface of the hotwell to the water level in the boiler is 12 ft. The end of the suction pipe is square cut, the total length of run 150 ft., and there are three globe valves and eight 90-deg. elbows. What head must the pump operate against, if the velocity is limited to 325 ft. per minute?

TABLE IV—FRICTIONAL EFFECT OF FITTINGS

Kind of Fitting or Cause of Resistance	Equivalent Length of Pipe to Be Added for Frictional Resistance of Fitting					
	1 In. Pipe	1¼ In. Pipe	1½ In. Pipe	2 In. Pipe	2½ In. Pipe	3 In. Pipe
90 deg. elbow	6	8	8	8	11	15
Long rad. bend	3	3	3	4	6	8
Return bend	12	16	16	16	22	30
Globe valve	6	8	8	8	11	15
Strainer or foot valve	12	16	16	16	22	30
Square entrance	6	8	8	8	11	15

Solution: The limiting velocity is $\frac{325}{60} = 5.4$ ft. per sec. Looking in Table III, we find a 1¼-in. pipe will discharge this quantity of water with a velocity of 5.4 ft. per sec., and is the size required. The friction head for 100 ft. run, with this pipe size and velocity is 11.9 ft., or $11.9 \times 1.5 = 18$ ft. for 150 ft. run. The friction due to the suction inlet is 8 ft.; from Table IV globe valves, $3 \times 8 = 24$ ft., and 90-deg. elbows, $8 \times 8 = 64$ ft. Pressure head due to the boiler pressure is $\frac{130}{0.433} = 300$ ft., making the total head pumped against, $18 + 8 + 24 + 64 + 300 = 414$ ft. of water column, or a pressure of $414 \times 0.433 = 180$ lb. per sq.in.

The power required for operating a pump is made up of two parts, that expended in lifting the water, and that in overcoming the frictional resistance of various kinds. This is given by the formula,

$$\text{b.h.p.} = \frac{W \times H}{33{,}000 \times E}$$

in which

b.h.p. = the brake horsepower required to operate the pump;

W = weight of water pumped, in pounds per minute;

H = the total head pumped against, which includes the equivalent pressure or measured head, velocity head, friction head;

E = the efficiency of the pump.

The value of E varies widely, even in pumps of the same kind, depending upon size, speed, physical condition and state of adjustment. For accurate work guaranteed figures should be obtained from the manufacturers for the particular pump under consideration and the conditions under which it is to operate. With boiler-feed pumps of medium size in the average well-kept power plant we may assume that for direct-acting steam pumps, E = 65 to 70 per cent; power pumps, E = 70 to 75 per cent, and centrifugal pumps, E = 60 to 65 per cent.

Assuming an efficiency of 70 per cent in case of the boiler-feed pump in the previous example, the brake horsepower required under maximum operating conditions would be

$$\text{b.h.p.} = \frac{(25 \times 8.3) \times 414}{33{,}000 \times 0.7} = 4 \text{ (nearly)}$$

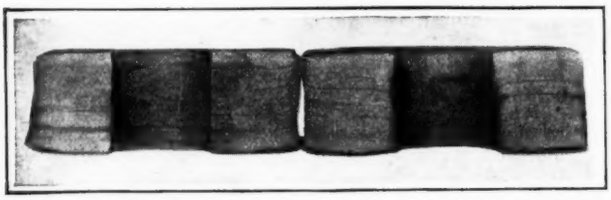


Fig. 1—Drilled hole (left) compared with punched hole (right)

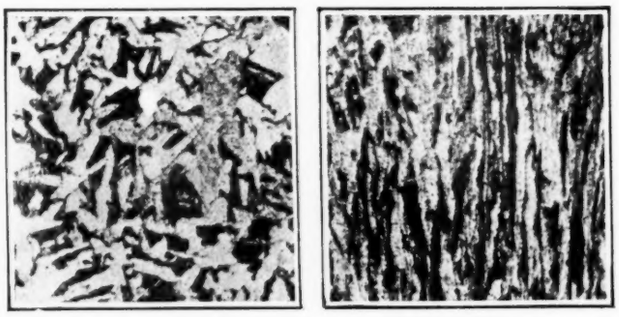


Fig. 2—Unstrained metal away from hole. Fig. 3—Brittle cold-drawn metal near punched edge

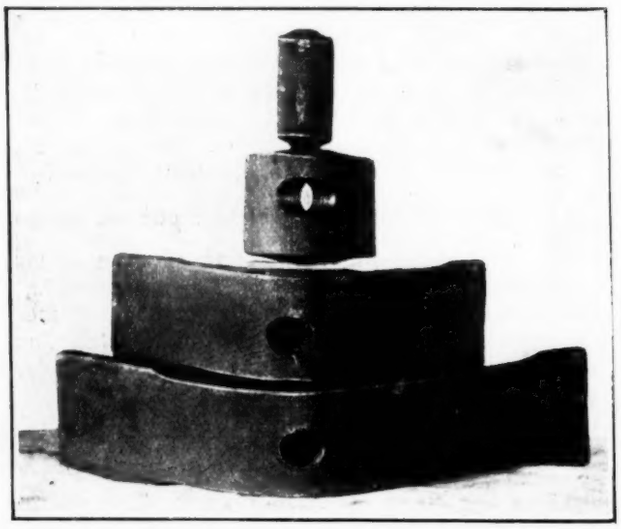


Fig. 4—Effect of kind of hole on bending (see text)

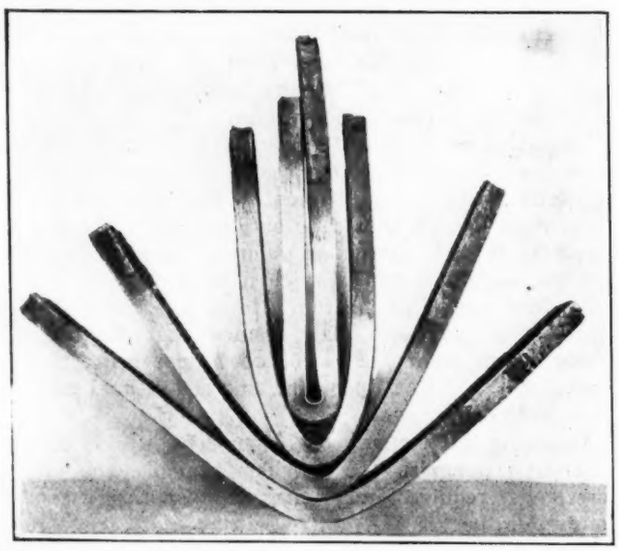


Fig. 5—Side view of test speciments shown in Fig. 4

Why A. S. M. E. Code Prohibits Punching and Shearing

WHILE punching and shearing of boiler plates are generally cheaper operations than drilling or shearing plus planing, the simpler methods are to be discouraged, particularly with thick plates, because of damage done to the metal.

The A. S. M. E. Boiler (construction) Code has always prescribed the planing of sheared edges to facilitate calking. The newly revised 1924 Code also prohibits the punching of rivet or staybolt holes full size. Paragraph 253, dealing with this point, runs as follows:

> All holes in braces, legs and sheets for rivets or staybolts shall be drilled full size with plates, butt straps and heads bolted up in position, or they may be drilled or punched not to exceed ¼ in. less than full size for plates over $\frac{5}{16}$ in. in thickness and ⅛ in. less than full size for plates not exceeding $\frac{5}{16}$ in. in thickness and then drilled or reamed to full size with plates, butt straps and heads bolted up in position.
>
> Such holes shall not be punched in material more than ⅝ in. thick.
>
> The finished holes must be true, clean and concentric.

This paragraph is based on the recognition of two facts: first, that punching always strains the metal adjacent to the punched edge, and second, that the penetration of the weakness increases with the thickness of the plate. The latter fact accounts for the requirement of more material to be drilled out with plates over $\frac{5}{16}$ in. thick and for the entire prohibition of punching for plates over ⅝ in. thick, a thickness that will be exceeded more and more frequently as steam pressure continues to increase.

A scientific investigation of the effect of punching and shearing on thick plates has recently been made by the Lukens Steel Co., through whose courtesy the information and photographs for this article have been obtained.

Fig. 1 shows two half-sections, one of the drilled hole (left) and the other of a punched hole. The sawed surfaces have been deeply etched with acid to show the effects of drilling and punching. The bad effects of punching are manifest.

The effect on the metal structure is further revealed by the microphotographs shown in Figs. 2 and 3. Fig. 2, taken about ⅛ in. away from the punched edge, shows the normal structure, while the effects of cold drawing immediately adjacent to the hole are apparent in Fig. 3. The grains have been elongated and pulled out of shape, as in cold-drawn wire, which is known to be extremely brittle.

To carry the investigation a step farther, bending tests were made on four test pieces taken from the same portion of the same plate. The first was left without a hole, the second was drilled, the third and fourth were punched. These were then bent in the manner to be described and arranged in the following order, from top to bottom, for photographing (see Figs. 4 and 5): Undrilled test piece, drilled test piece, punched piece bent with fin inside, punched piece bent with fin outside. The undrilled piece was given the ordinary cold bending test, while the others were bent slowly until cracks developed. The results shown by the photographs tell their own story. The plates shown are all ⅝ in. Similar comparative results were obtained with other thicknesses, although the thinner plates naturally withstood more bending before cracks developed.

Wear of Reduction Gear Teeth Measured From Plaster Casts

By H. B. CHAPMAN
Manager Steam Marine Division, Westinghouse Electric & Manufacturing Company

Tells how an accurate cast of a helix or tooth may be easily made and used for measuring thickness. If taken at regular intervals, these casts will represent a permanent record of reduction gear wear.

NO MATTER how experienced an engineer may be, he cannot by visual inspection judge with any degree of certainty the wear that has taken place on the teeth of gears. Any records made by such a method would at best be of very general interest and would be of no value for engineering or design purposes. In fact, it is hard to imagine how they could be used at all.

Desiring to keep a complete record of the life of reduction gears on turbine-driven ships, the Westinghouse Service Department developed a simple and inexpensive yet thoroughly reliable method of making permanent records of the condition of the gear teeth. Not only is it possible, after a short acquaintance with the system, to foretell the approximate life of the gear, but some interesting facts were brought to light on the subject of undercutting and pitting.

The requirement for a satisfactory record is an exact full size reproduction of a section of the gear at the time the record is made. The method employed in making the artificial stone cast is practically identical to that used by dentists in making casts for plates of artificial teeth. The negative impression is produced with paraffin wax. The better grade of wax, such as made for domestic purposes, is best, being harder and having a higher melting point. We have found wax by the trade name of "Gulfwax" very good.

The teeth should be wiped clean and dry and marked permanently, so they can readily be located when it is desired again to take a cast. A dam built around the sections to be molded should be parallel to the face of the teeth and about 3½ in. wide by 5 in. long. For this purpose ordinary putty mixed fairly stiff works very well. The dam should extend ¾ in. above the top of the teeth with sufficient melted wax poured in to fill the dam completely.

When the wax has become quite solid, the putty can be removed and the cast pried up with a penknife from the bottom of the teeth. The wax negative is then mounted on a piece of board a little larger than itself and the same putty used to make the sides of the plaster mold.

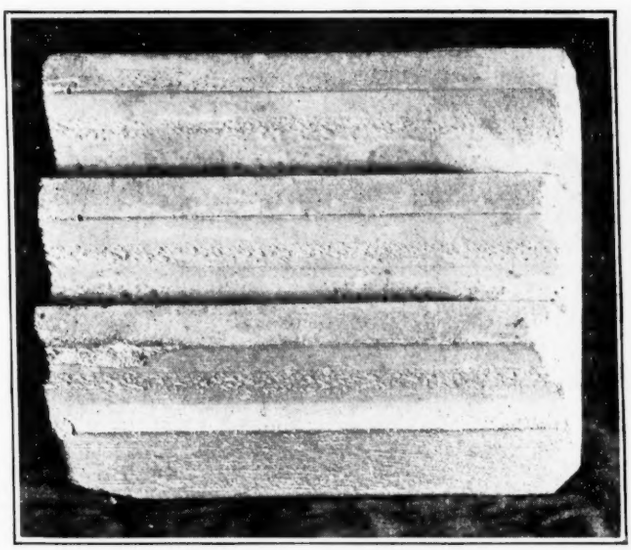


Fig. 1—Cast of driven side, forward marine helix

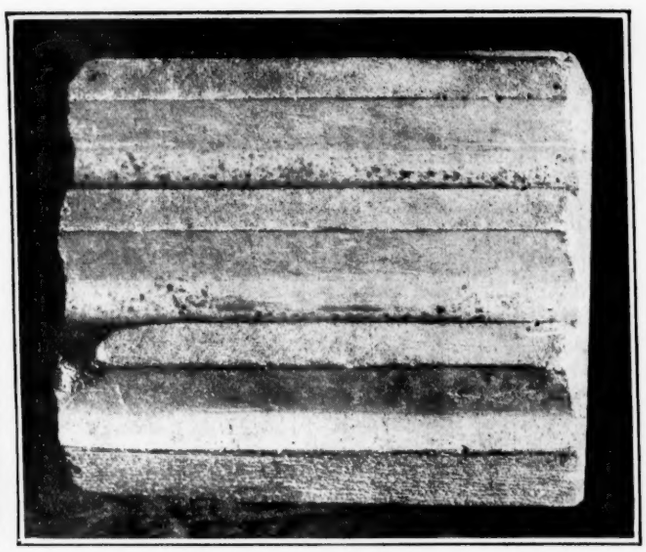


Fig. 2—Smooth idle side of same cast as in Fig. 1

A preparation used by dentists, called "Spence's plaster," which is a refined portland cement, proved suitable for this work. It is sold by most dental supply houses at about $2 for a nine-pound carton. This is mixed, preferably in a dentist's rubber plaster bowl, by adding the plaster to water, continually joggling the bowl to exclude the air bubbles. The cement should be mixed quite thick, in fact like soft putty, and then pieces about one inch long are taken up, rolled in the hand and laid between the teeth in the center of the wax cast. By continually rapping the cast on the table, the plaster will spread the full length of the teeth. The gradual filling of the mold in this way will exclude all the air bubbles and pockets.

We also found that the shrinkage in the wax negative gives a slightly larger plaster cast, which in turn shrinks when thoroughly dried out to practically the identical size of the tooth, permitting the use of a tooth gage on the cast, which shows what is going on very clearly. The plaster should be allowed to set at least

six hours, the wax melted out or pried off, and the cast dressed up with a hacksaw and file.

At present all necessary information concerning the gear is written on a label which is affixed to the base of the cast, the whole base being then given a coat of waterproof varnish. A sample of the class of information required on the label of a marine gear cast is given in the table for the guidance of those who may see an application of this recording system in other fields.

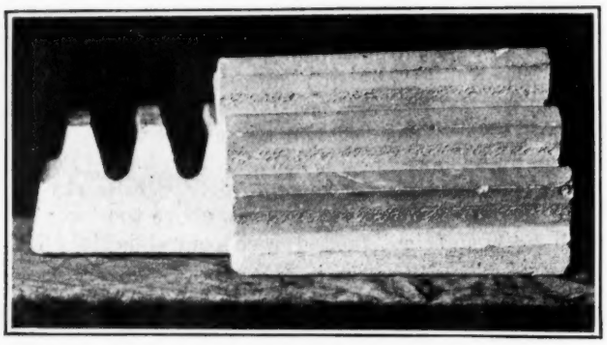


Fig. 3—Driven side of helix shown in foreground

CAST OF S. S. MOUNT CARROLL

Main gear	Forward helix
Impression taken	May 26, 1923
Thickness at date	0.446 in.
Addendum	length, 0.286 in.
Thickness when new	0.448 in.
Operation to date	130,000 miles

Although this method was originated for marine use, it is applicable to any kind of gear teeth. Figs. 1 and 2 are both from the cast of a forward marine gear helix.

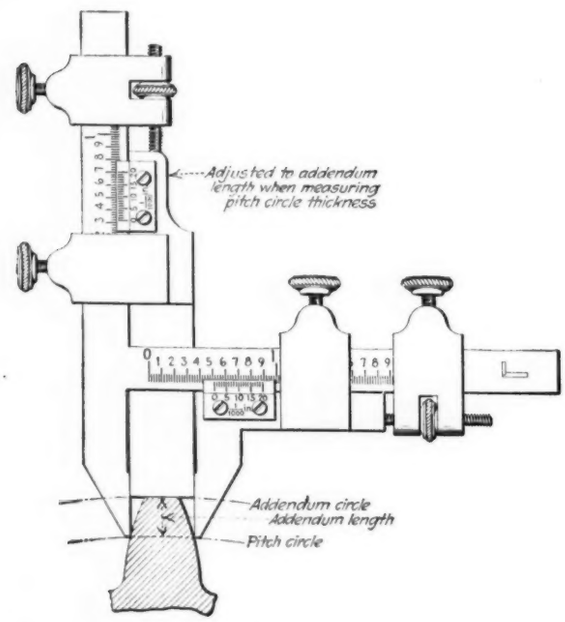


Fig. 4—Thickness measured at pitch circle when addendum length is known

The wear is plainly indicated on the driven side of the tooth cast, Fig. 1, and the smooth surface is evident on the idle side, Fig. 2. The driven side of the after gear helix appears in the foreground of Fig. 3. Tooth thickness may be measured as at Fig. 4, where the measured thickness is preferably at the pitch circle. The addendum length must be known so as to adjust the caliper for taking this thickness. If not known, some standard length must be chosen and recorded, so it can be used in measuring subsequent casts.

A study of a number of gear impressions brought out the fact that apparently much of the wear is a flaking action, which when it becomes intensive, causes a continuous depression below the pitch line, termed undercutting. The blow delivered by the pinion tends to crystallize the metal which receives the blow and it finally flakes out, giving at first glance a deceptive idea of the amount of wear that actually is taking place. It was also found that this wear falls off gradually while the gear is in service, finally ceasing altogether, leaving a gear the useful life of which will exceed the life of the ship in which it is installed.

A series of records, taken at intervals of three or six months, might at first lead the observer to believe that the gear was doomed for the scrap pile within a year. But it was found that ships having approximately a half-billion miles to their credit, such as the S.S. "Maui" (400,000 miles at last inspection) and the S.S. "Goloa" (250,000 miles at last inspection), show less wear as time goes on. A curve made from the gears on one ship appears in Fig. 5.

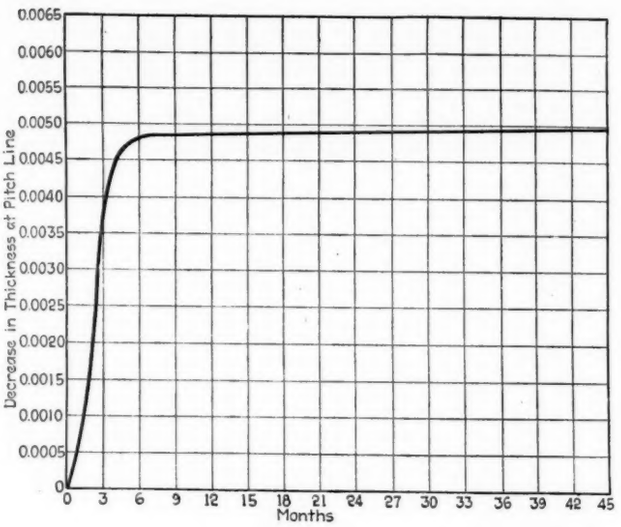
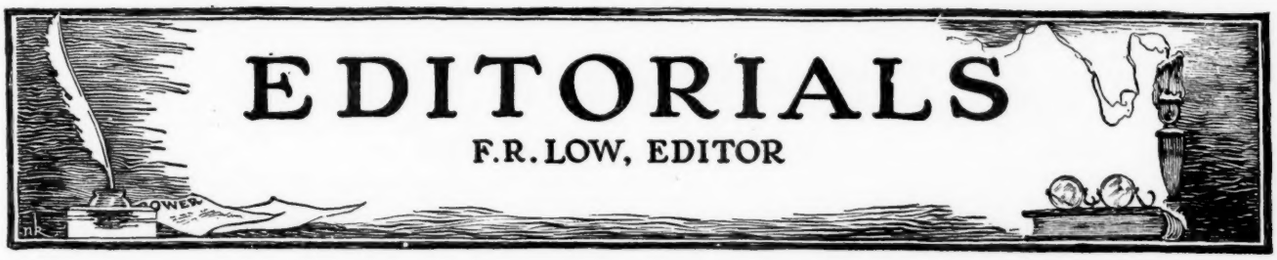


Fig. 5—Tooth wear practically stops in six months

An interesting article in the November 14 issue of the *Engineer* (London) takes up two commonly held fallacies in connection with the use of superheated steam. Both of these points have been discussed editorially in *Power* and it is interesting to note that the writer in the *Engineer* takes the same stand in both cases. The first conclusion arrived at, after a careful analysis of a specific case, shows that thermodynamic considerations alone do not prevent the bleeding of steam in the superheated region for heating feed water. Such bleeding shows an actual gain, at least in the theoretical case. The other point brought out is that there is no reason to expect a reduction in capacity per square foot of a heater supplied with superheated instead of saturated steam, at least under ordinary commercial conditions.



Supporting Research Work

WHILE there undoubtedly is, in all lines, a considerable amount of research work being carried on in the laboratories of technical schools, the full capabilities of these institutions are not utilized. In place of the common practice of several firms carrying on individual investigations on a particular subject, equal if not better results would be obtained if this work was centered in a single laboratory and funded by the firms engaged in the particular industry.

In the development of the oil-engine probably thirty firms are engaging in investigations covering such lines as atomization, penetration and phase of combustion within the solid-injection engine. At the same time the Penn State College is engaging in similar research work. It would seem that if the industry would support the college laboratory at an outlay of one-half or even one-fifth the present total cost of the simultaneous investigations, more data of great accuracy would be obtained. Probably the habit of manufacturers' preferring to do their own research work is based on a desire to discover something withheld from competitors. Unfortunately, discoveries have a way of wandering from home, so that it is, indeed, an unusual organization that can keep its news within its family walls. Greater support of college research laboratories would show a return far out of proportion to the money invested.

Asphyxiating the Power Plant

IT SEEMS to be an inherent condition in many plants that no arrangements have been made to provide a definite channel or route that will assure at all times an adequate supply of air to the fuel bed. There are exceptions to this, of course, notably in the larger forced- and induced-draft plants and where air warmed by the losses of the generators is delivered to the fuel bed. In the natural-draft plants, however, the air is allowed usually to get into the furnace as best it can.

During the summer months this situation does not matter particularly; doors and windows are thrown open wide. In the winter it is a different story. Doors, transoms and windows are closed up tight to make the boiler room habitable, so that the supply of air to the fuel bed may be so curtailed as to interfere seriously with the capacity of the plant, particularly when overloading occurs and where stack capacity is barely sufficient.

When entering a boiler room from the outside during the winter, one frequently encounters an appreciable vacuum of such degree that considerable physical effort is required in closing the doors against the outside pressure. This is a definite protest for a more liberal air supply, yet, strangely enough, it is usually passed over unnoticed by the boiler-room force. A draft reading showing the pressure difference between the inside and outside would be surprising, and when interpreted in terms of stack height or other equivalent, would emphasize the need for more careful attention to this small but important detail.

Therefore, for those plants where such a condition can exist, and in all plants upon general principles, it will be well to look to the air supply. The subject is broader than appears at first sight, as it involves matters such as moisture, dirt, débris, ventilation and the prevention of condensation, where the turbine or engine room is adjacent to the boiler room. Since every degree increase of the incoming air is a direct gain, and as often much warmed air is available, it is surely worth while to use it.

Protecting Power Equipment

IF IT were possible to compute the total loss to industry due to failure to protect power equipment properly, it would run into a figure each year of no small proportions. Such losses include not only the cost of repairing or replacing damaged apparatus, but also that due to loss of production and employees' wages, on account of having equipment out of service. These interruptions are caused not only by failure of equipment that drives the industrial apparatus, but also by breakdowns in the plants that supply the power to electric motors. Some of these losses are unavoidable, but a large percentage can be prevented by proper protection to equipment and better maintenance methods.

The question of how far protection of power equipment should be carried is one of economics, the answer to which must be found in what protection will save in dollars and contribute toward rendering better and more reliable service. In this respect it is possible to load equipment up with so much protective apparatus as to be a detriment to service rather than a benefit. However, this is the exception and not the rule.

Too often the use of safety devices is looked upon from what they cost rather than from what they may save. Protective apparatus should be considered much as insurance against loss of equipment. It is a question of how much it is worth to carry the responsibility of the losses they may prevent in damage to equipment and interruptions to service.

Not infrequently, machines are found, the cost of which runs into large sums, installed without suitable protection, when this protection could be supplied at a trifling expense. Not so long ago a two thousand-kilowatt turbine and alternator were wrecked owing to the lack of a reverse-current relay in the generator circuit. When the cost of the protective equipment is compared with that of the turbine and generator, it is difficult to see why those responsible for the installation failed to provide the relay. This is only one of many failures that have occurred to power equipment due to lack of protection, but it is representative.

There is the argument that the operator may put too much dependence in safety devices if he knows they will do the things that he might be required to do otherwise. This, however, is a matter to be taken care of in operating procedure. Because the operator on an elevator may depend upon the limit stops at the terminal landings to stop the car, is no reason these limits should be taken off with the risk of wrecking the machine and injuring the passengers. It would be unwise to remove the overspeed stop on a turbine or engine just because the operator might be inclined to give less attention to the condition of the governor, believing that the machine would be stopped automatically should it begin to race. The ideal combination is adequate protection combined with responsible operators.

The Electric Heat Fallacy

A CIGAR may be lighted conveniently and elegantly with a dollar bill. Houses may be heated from cellar to attic by electricity in a manner to suit the most fastidious. Only one objection may be raised to either of these propositions; they are intrinsically extravagant. Yet, statements recently sent broadcast by the daily press imply that electricity will some day be the prevailing method of heating houses. This being an "age of electricity," it is only necessary to mention some new applications of electricity to excite the interest of the average citizen. Electricity drives trolley cars, elevators and trains; it lights our streets and houses—all with great advantage and economy. Why not heat all houses by electricity and do away with ashes, dirt and trouble?

The householder can hardly be blamed for ignorance of certain fundamental engineering considerations that strictly limit the application of electric heating. Engineers, however, should be careful not to raise false hopes concerning such matters. Electric heating has many worth-while applications and their number will doubtless continue to increase, but the application of electricity to the heating of entire houses is, and must continue to be, an unthinkable extravagance except under extraordinary circumstances where enormous expense is no deterrent or where electrical energy from running water is available and would otherwise go to waste.

Take a typical house in the northern half of the United States, consuming, say, ten tons of thirteen-thousand-B.t.u. coal per year at a cost of sixteen dollars per ton. The yearly fuel cost is one hundred and sixty dollars. Assume that the efficiency of the heating system is fifty per cent, a figure easily attained with unskilled operation. The equivalent amount of electrical energy, applied with one-hundred per cent efficiency, is about seventy-five thousand kilowatt-hours.

At the present average household lighting rate of eight cents per kilowatt-hour this would cost six thousand dollars per year. At the low industrial rate of one and one-half cents it would cost over eleven hundred dollars. It would still cost seven hundred and fifty dollars per year at the extremely low rate of one cent. Remember this is not for a mansion, but for the average house, owned by the average man of small salary!

In the best central stations of today at least seven pounds of coal must be burned at the station to produce the heat equivalent of one pound at the customer's home, or three and one-half pounds allowing fifty per cent efficiency of coal utilization in the home. It is elementary economics that the user of electric heat must pay for this coal. In addition, even where the heating load is strictly "off peak," there is still a necessary burden of operating costs and profits.

Another way to look at it is in terms of the equivalent cost of coal. Comparing electricity, perfectly utilized, with thirteen-thousand-and-six-hundred-B.t.u. coal used with fifty per cent efficiency, each cent per kilowatt-hour is equivalent to forty dollars per ton for coal. At two cents per kilowatt-hour electricity is equivalent to eighty dollar coal, at four cents to one hundred and sixty dollar coal and at eight cents to three hundred dollar coal.

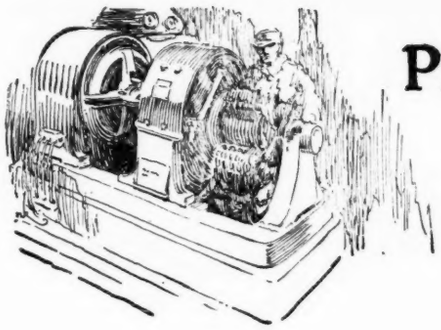
The problem under discussion is the general heating of entire houses by electricity. The facts cited are not inconsistent with known facts regarding the occasional use of waste hydro-electric power for the heating of entire houses or the general use of localized electric heat in household and industrial appliances where the efficiency and convenience of application may outweigh the much higher unit cost of electrical energy.

Making Building Material from Fuel

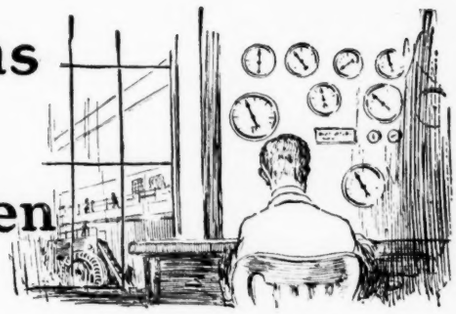
IN FORMER days the bagasse, or the fiber of cane after the removal of the sugar juice, when burned under the boilers, was ample to supply all the steam necessary. As crushing methods improved, the percentage of bagasse decreased until it was necessary to supply half the fuel requirements by the purchase of oil or coal. Recently, the cane fiber has been put to a new use of such a commercial value that its consumption under boilers will be entirely discontinued. In Louisiana the waste is baled much after the fashion of hay, and is then freighted to a factory where it is converted into a building composition board. The value of the bagasse as a building material is such that a mill can no longer burn it.

Still another step in making better use of byproducts now used as fuel is in the lumber mills. At each mill of any pretention the waste lumber has been so great that all of it could not be consumed under the boilers. The remainder was dumped into a huge vertical steel sheet, often brick-lined, and so destroyed by burning. It is a common sight to see the long flaming shaft from a distance of twenty or thirty miles. Science has stepped in to put a stop to this wanton destruction of valuable material, and has discovered a method of converting the waste lumber into paper. While at present only the lumber not needed for the boiler furnace is put into the paper mill, the value as paper is so much greater than the fuel value that it is only a question of time when the boiler plants will discontinue the burning of lumber and use coal or oil, and no doubt open a new field for the oil engine.

With the Ford offer withdrawn and with President Coolidge, Secretary Weeks and Senator Underwood arriving at an understanding, passage by the Senate of the revised Underwood bill on Muscle Shoals at this session of Congress seems assured. The bill has still to pass the House, however, where opposition is expected and where it may be delayed or defeated, and the next Congress, which will meet in December of 1925, unless a special session is called, will then have to begin at the beginning and go all over it anew.



Practical Ideas *from* Practical Men



One Way of Reducing Maintenance Costs

The cost of maintenance work could be reduced considerably if the workmen were trained in the art of conserving material. There seems to be a great tendency among pipefitters to break a fitting whenever it is necessary to change a small pipe connection on the line. This is often done unnecessarily. Breaking the fitting with a hammer, even though a second hammer is held behind the fitting, not only destroys the fitting but often causes strains that result in leaks in other parts of the line. Not infrequently a Stillson wrench is used on lead piping, resulting in the breaking of the pipe.

Small brass valve bonnets are often chewed up with a pipe wrench, and it is a common habit of pipefitters to screw a valve on a line with the wrench on the wrong end of the valve body. I have even noticed fitters screw the end of a pipe into the body of a valve until it touched the web, which held the seat simply because an extra long thread had been cut on the pipe.

It would be possible to go on and name many other things that are simply the result of not caring, some of which often result in seriously tying up of certain machines about the plant. Much of this could be avoided if workmen were more thoroughly trained and instructed in the proper use of tools and the conservation of materials.

RAYMOND L. DREW.

Philadelphia, Pa.

Why Do the Indicators Give Different Diagrams?

The illustrations herewith show a pair of indicator diagrams from the steam cylinder of an ammonia compressor. Previous to taking the diagrams there was a heavy knock as the crank went over the center away from

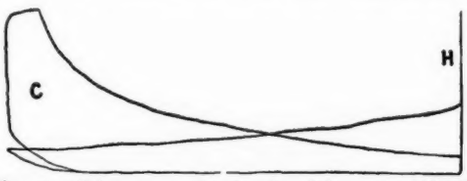


Fig. 1—Diagram obtained with a practically new indicator

the cylinder, but in going over the center nearest the cylinder there was no knock and everything seemed to be all right. In order to determine readily the cause of the knock, we put on a practically new indicator and got a diagram as shown in Fig. 1. Next we removed the steam valves and found the valve on the head end out of place about ¼ in. This was set according to the marks and the cover replaced. As we got a fairly good diagram for the crank end, we left that end alone. Then we took diagram No. 2, which was the same as No. 1 in every respect, showing no improvement on the head end. After that we turned the piston rod back toward the head end and the knock went out of

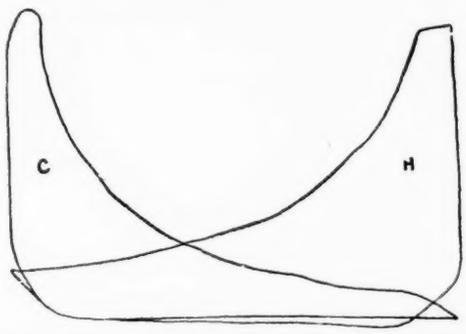


Fig. 2—Diagram obtained with a different indicator with compressor operating under the same condition

the machine, but still we continued to get the same kind of diagram. We finally concluded that the indicator was at fault, so we got another indicator and put it on and obtained the diagram shown in Fig. 2.

Now as far as we could see there was no difference in the action of the two indicators. We used a 60-lb. spring with 120 lb. steam pressure, and at the time the diagrams were taken the compressor was operating with 120 lb. condenser pressure and 14 lb. gage pressure on the suction.

Probably some reader can tell us why we do not get as good a card with one indicator as we do with the other one under the same conditions.

Rochester, N. Y. R. G. SUMMERS.

Changes Made in the Power Plant Paid One Hundred per Cent Dividends

There are many power plants in the paper industry where a few thousand dollars spent on improvements will earn yearly dividends as high as 100 per cent. I know of one instance where this was accomplished, and undoubtedly it could be done in many other plants. The plant in question had a 26x42-in. simple, non-condensing Corliss engine and two 200-hp. waterwheels, all belted to the mainshaft. There were also two 60-hp. slide-valve engines driving the paper machines. The boiler plant consisted of three water-tube boilers of 2,000 sq.ft. heating surface each designed to operate at 175 lb. pressure. The boilers, however, were operated at 125 lb., as the cylinder of the main engine had been rebored so many times that it was not safe to increase the pressure on it. A portion of the exhaust from this engine was used to heat the mill in winter, but only

one-third was required, and in summer none of the exhaust was needed. The two small engines exhausted into the driers at 8 lb. back pressure, and any excess over the drying requirements went to the feed-water heater. These engines developed a total of 120 hp. and used slightly over 5,500 lb. of steam per hour. The main engine carrying a load of 500 hp. used 11,500 lb. per hour, and the total boiler load was around 17,000 lb. per hour.

The management decided to replace the old water-wheels with a new hydro-electric plant and run the beater room with individual motors. The cost of the new hydro-electric equipment was about $80,000, and the power derived from it was around 550 hp. This was 150 hp. more than the old wheels developed, due to improved design. This resulted in a decreased load on the engines and lower fuel consumption. The fuel saving was estimated at $9,000 per year. It is not the purpose of this article to describe in detail the design and operation of the hydro-electric plant, but it is simply mentioned to show the comparative savings made by this method and the greater saving made by improving the steam plant. The owners made no objection to spending $80,000 on a project that paid dividends of 11 per cent on the investment, but it was a difficult task to persuade them to invest a few thousand dollars in the steam plant, even though the entire cost of the improvements would be returned in one year.

Separate Engines Form Compound Unit

After the new hydro station was put in operation, the load on the main engine was reduced to 350 hp. The engineer in charge advised that the power to drive the paper machines should be generated by the main engine and the small engines be replaced with motors. The problem was complicated somewhat by the necessity of supplying the paper machines with 5,500 lb. of steam per hour at 8 to 10 lb. pressure. To have the large engine operate against that back pressure would not be economical, since only half of it could be used by the driers. With the engine generating power for the paper machines the load would be increased by 120 hp., giving a total load of 470 hp. and a steam consumption of 11,000 lb. per hour exhausting at atmospheric pressure. Since all of the exhaust could not be used even during the heating season, the engineer proposed the purchase of a second-hand 16x42-in. Corliss which was available. This would be installed to carry part of the load and operate at 175 lb. steam pressure. Part of the exhaust would be used to supply the driers and the remainder would be utilized by the larger engine, which would operate condensing and act as a low-pressure cylinder. Thus the advantages of a compound condensing engine would be secured and the economies of utilizing bled steam would be realized.

The power that would be developed by the 16x42-in. engine with a pressure of 125 lb. and operating at the same speed as the large engine (80 r.p.m.) was roughly estimated by the engineer to be 189 hp. and would have a steam consumption of 4,347 lb. per hour. But if the engine operated at 80 r.p.m. at 175 lb. pressure, the amount of steam used by this engine would be about 5,800 lb., and assuming that the receiver pressure would be maintained constant at 10 lb., the power output would be around 252 hp. The engineer figured that at least 50 per cent more steam would have to be passed through the high-pressure engine since 5,500 lb. would be bled, and to do this and still keep the cutoff at an economical point, the engine would have to operate at 120 r.p.m. It had formerly run at 125 r.p.m. so no difficulty would arise from operating at this speed. As the 5,500 lb. of steam would develop about 225 hp. in the high-pressure cylinder, this left an additional 245 hp. to be developed by steam used in both cylinders and expanded to 26 in. of vacuum. He figured that the two engines operating as a unit would develop a horsepower on 13 lb. of steam and would require 3,185 lb. of steam in addition to the bled steam. Therefore, the total steam entering the high-pressure cylinder would be 8,685 lb. per hour, which would represent the total steam consumption of the mill.

Two Engines Operate as One Unit

Now on account of the greatly increased volume of steam at 10 lb. pressure, the low-pressure cylinder must operate at a higher piston speed to use 3,185 lb. per hour and develop the necessary power. This engine was speeded up to 96 r.p.m. and the pulley on the mainshaft changed to compensate for the increase in speed of the flywheel. The cutoff had to be made later than it had been fromerly, and the valve setting was changed considerably, as much less compression was needed. Under normal operating conditions the high-pressure engine developed between 340 and 350 hp., while the low pressure developed between 110 and 130 hp. No trouble was experienced with the speed regulation, and owing to the large volume of the driers (nearly 1,000 cu.ft.) the receiver pressure did not vary much.

During the winter the engines operated as a compound non-condensing unit, and with the exception of a small amount of steam bled to the heater, all the steam passed through both cylinders and was exhausted from the low-pressure engine at 8 lb. back pressure to the driers and heating systems. The steam rate under these conditions was about 20 lb. per hp. giving an average steam flow for the year of 9,043 lb. per hour.

To determine the saving made by this plan, we must first deduct the coal saved by improvement made in the hydro-electric plant. The steam plant originally burned 25 tons of coal in 24 hours and used 17,000 lb. of steam per hour. This gives an evaporation of 8.16 lb. of water per pound of coal. The engine load was reduced 30 per cent by the new hydro plant, thus saving 3,450 lb. of steam per hour or 422 lb. of coal. The plant operating 24 hours a day, 300 days a year and with coal costing $6 a ton, the saving amounted to $9,115 on an $80,000 investment. The boiler load was then 13,550 lb. per hour, and the changes in the engine room reduced this to about 9,043 lb. per hour, or a saving of 4,507 lb. of steam per hour was made by remodeling the steam equipment. This is equivalent to 552 lb. of coal per hour, or a yearly saving of $11,923, which more than covered the cost of the additional engine and accessories.

No estimate is made of the steam used by the condenser pump and the boiler-feed pump as they exhaust into a closed feed heater (No. 2) at 30-lb. back pressure, raising the feed water to a final temperature of 260-265 deg. F. The condensate from the driers and makeup water from the condenser enter a hotwell tank and then pass to the No. 1 heater, where the temperature is raised to 220 deg. by bled steam and condensate from the No. 2 heater. It might be said in conclusion, that this arrangement gives a heat balance equal to that being obtained in many other newer plants.

Holyoke, Mass. A. F. Sheehan.

Comments from Readers

Water-Column Connections

I was much interested in the inquiry of J. A. B., entitled "Water-Column Connections," which appeared in the Nov. 18 issue.

Feed-water regulator connections taken from water-column connections, as stated in the answer, have been found satisfactory. It is my experience that the best results are obtained by making the regulator connections as close to the boiler as possible. This eliminates any tendency of the pressure drop that occurs in the column when it is blown down from adversely influencing the operation of the feed-water regulator at blow-down periods. This information may be of assistance to other readers who are operating feed-water regulators piped to the water-column connections.

Erie, Pa. G. J. COPES.

Turbine Performance from Steam Quality Measurements

In his article "Turbine Performance from Steam Quality Measurements" (*Power*, Oct. 28), N. L. Johnson contends, in effect, that the total heat per pound of steam supplied to the prime mover minus the heat per pound of exhaust steam is equal to the heat converted into work per pound of steam and is therefore a measure of the water rate of the prime mover. This contention is absolutely correct for all practical purposes. He is also correct in his statement that the idea is not new, but has not found wide employment. In fact, I have found engineers of experience startled by the idea and doubting the correctness of it, when I proposed a method for testing steam prime movers, somewhat similar to that described by Mr. Johnson.

I regret exceedingly the necessity for differing from Mr. Johnson in the interpretation of that thermodynamic fact. I also consider the example Mr. Johnson selected for demonstrating the application of his very useful method rather unfortunate, for in the case considered the use of his method has little, if any, advantage over other methods mentioned in the beginning of his article, particularly when the exhaust steam is below atmospheric pressure and contains considerable moisture.

The statement that the difference between the heat contents of the steam at the nozzles and that at the exhaust is equal to the heat converted into work is correct only if by work the *input* of the prime mover is understood. To say that it is equal to the work at the coupling of the prime mover is equivalent to the assumption that the prime mover has an efficiency of 100 per cent, an assumption which is hardly permissible, even in work of a very approximate nature. The efficiency of a prime mover driving an electric generator is usually less than that of the generator, and to neglect the first one, while the second is given due consideration, is rather inconsistent. Consequently, Mr. Johnson's figure of 10.64 is not the pounds of steam required to produce a horsepower-hour at the coupling (*output*), but the pounds of steam per horsepower-hour *input*. To get the steam consumption per horsepower-hour at the coupling it is necessary to take into consideration the mechanical efficiency of the prime mover, which is probably about 95 per cent (from the manufacturers' data) at that load. The steam consumption per horsepower-hour at the coupling is therefore, $10.64 \div 0.95 = 11.20$ lb., and the steam consumption per kilowatt-hour is $\frac{11.20}{0.746 \times 0.94} = 15.97$ lb. (say 16 lb.).

Had Mr. Johnson considered the case of a reciprocating engine exhausting atmospherically steam containing considerable moisture, the advantages of his method would have been more apparent, as in that case it is practically the only method available for frequent use. His answer of 10.64 lb. would have been the steam consumption per *indicated* horsepower-hour. To get the steam consumption per *brake* horsepower-hour (at the coupling) the mechanical efficiency of the prime mover would have to be given due consideration. Both efficiencies must be considered in figuring out the steam consumption per kilowatt-hour.

The power-house operator is vitally interested in keeping continuous watch on the steam consumption of his prime movers. Cases where the steam consumption is 60 per cent more than the guaranteed amount are not infrequent in small plants where the prescribed operating conditions with respect to steam pressure, steam superheat, vacuum, load, etc., are not maintained. Particularly is the steam loss great when the valves and pistons are not tight and the steam leaks into the exhaust. This condition is not noticeable and is only guessed at by the experienced operating engineer. But the management very often does not see the necessity for overhauling so long as the unit is operating "satisfactorily."

The A.S.M.E. Power Test Code has provided admirably accurate tests, but they are too expensive for frequent use. To detect material increases in the steam consumption promptly, daily test should be made, and to make such tests practical they should be simple and inexpensive.

The manipulation of the Thomas calorimeter requires considerable skill, and the computations are somewhat involved for the busy operating engineer. Hence the use of it should be limited to cases where no simpler method is available. It is not apparent from the article how a sample of exhaust steam at 2 lb. absolute pressure and carrying about 15 to 18 per cent moisture is induced to enter the Thomas calorimeter, and I am not certain that it is practicable. Assuming that it could be done, wherein is the advantage over the use of a flow meter, if the use of the latter is possible? Certainly not in the relative simplicity, inexpensiveness and accuracy of results.

In my opinion, based on experience in testing steam prime movers at frequent intervals and continuously, the method to be used is dependent upon the type of prime mover and the operating conditions. The differ-

ent methods and the conditions they are to be used under are about as follows:

1. Turbines operating non-condensing or condensing into a surface or jet condenser.—Use a flow meter at the nozzles. (Where surface condensers are used, it may be found advantageous to use a hot-water meter in the condensate return line in place of the flow meter or to check the flow meter.)
2. Reciprocating engines:
   (*a*) Operating non-condensing.—Use Mr. Johnson's method under discussion.
   (*b*) Operating condensing into a surface condenser.—Use a hot-water meter in the condensate return line.
   (*c*) Operating condensing into a jet condenser.—Use F. S. Youtsey's method (*Power*, June 17, 1924), or the somewhat similar heat-balance method used by the writer.

With the heat-balance method the water rate of the prime mover, the ratio of the amount of injection water to the steam and the tightness of the valves and pistons may be ascertained in less than two minutes at no expense whatever, except the initial cost of installing the weir. If a recording weir meter equipped with a recording differential thermometer is installed, a continuous record of the water rate of the prime mover may be kept.

E. OGUR.

New York City.

[Mr. Ogur has done a service in calling attention to some of the limitations of Mr. Johnson's method, but it appears that he has himself fallen into a few minor errors. The ninth paragraph of Mr. Johnson's article contains the following statement: "All this difference (between the heat in the steam at the throttle and at exhaust) must appear as power at the turbine coupling except the small part that forms the *external* losses due to gland leakage, friction of bearings and governor and radiation from the casing." The accuracy of this statement seems beyond question, but Mr. Ogur objects with some reason to the neglecting of these losses as immaterial. However, to avoid one error he falls into another. It is not exactly true that the heat drop is equivalent to the indicated horsepower (in case of a reciprocating engine). Nevertheless, it is only necessary to assume a suitable, though fictitious, value for mechanical efficiency, to make everything turn out well. —Editor.]

How Much Does the Water Level Drop in Different Types of Boilers?

The figures given by Prof. E. H. Lockwood in his solution of the problem, "How Much Does the Water Level Drop?" in the Nov. 25 issue, although theoretical, will obtain in practice under certain conditions, but in general the resultant figures are much too low.

The question, as asked, is of too broad a nature to give a specific answer. In order to answer the question definitely, certain conditions must be stipulated. Rise or fall in water level must be plotted against the curve resulting from change in rating, plotted against time.

Factors such as furnace design, kind of fuel fired and in the case of solid fuel the amount of fuel on the grates, affect the rapidity with which the rating changes when the air supply is shut off. Another factor that I have known to affect the rise and fall over 100 per cent when firing conditions were the same, is that of the boiler-water concentration.

The following approximate changes in water level due to change in rating have come under my observation:

In boilers of the Scotch marine type and oil fired, the following example is given: A boiler 16 ft. in diameter and 18 ft. long, with no refractories in the combustion chamber or furnace, and steaming at 150 per cent rating, gives a drop of at least 8 to 10 in. under these conditions.

Another example of extreme variation in water level is that of waste-heat boilers in a gas-making plant. Here we have the boiler operating at its maximum output, for a given installation, for approximately two minutes, and for the next two or three minutes there is no heat being delivered to it.

Rise and Fall Varies with the Boiler-Water Concentration

In one plant where I was assistant chief, we were operating on a four-minute cycle. When a boiler was cut in after cleaning and filling with fresh water, the rise and fall in level amounted to 8 to 10 in. The feed was treated with soda ash, and as the soda concentration of the boiler water increased, the rise and fall increased. By the end of three to four weeks the rise and fall had reached a point where the water alarm alternately blew for high and low water. This meant a rise and fall of from 26 to 30 inches.

It occurs to me that a design of waste-heat boiler, for use in plants where the heat is supplied intermittently as in gas making, employing some refractory to take up heat during the time the gases are passing through the boiler and giving heat to the tubes during the time the gases are shut off, would give a more steady steam supply.

In power-plant work I have seen this rise and fall in water level cover the limits of the gage glasses which vary from 18 to 22 in. in the particular plant I have in mind.

The more important question is, "How Much Does the Water Level Rise" when the rating is suddenly increased 200 per cent? If this rise is sufficient to carry water over to the turbines, main steam-line joints will suffer and accidents of a serious nature may result.

If boilers are banked in an emergency, the feed should be shut off as soon as water shows in the gage glass. If this is not done and the boilers are jumped from banked condition to 200 per cent or more of rating, water may be carried over into the steam mains.

If the air supply is shut off from an underfeed stoker-fired boiler, the drop in rating plotted against time will give a curve that starts out very steep, gradually rounding and flattening out, due to the stored heat in the fuel bed and brickwork.

Subsequent data will be submitted, giving the results of actual tests showing drop in rating and increase in rating plotted against time and showing fall and rise in water level. The salinity and alkalinity of the boiler water will also be determined. Other factors affecting the results will be incorporated in the report. An attempt will also be made to get some reliable data on the so-called express type of boiler, in which the rise and fall under given conditions will probably exceed that of any other type.

H. H. BAUMGARTNER,
Tech. Asst. Public Service Elec. & Gas Co. of N. J.
Newark, N. J.

The Protection of Pressure Gages

Having had considerable experience with gages on high-pressure hydraulic systems, I should like to make a few comments on the article by A. C. Williams in the Nov. 18 issue, entitled "The Protection of Pressure Gages." I have never noticed that ordinary machine vibrations were detrimental to a gage, as claimed by Mr. Williams. Neither will a fluctuating pressure prove injurious unless the fluctuations are rapid and allowed to continue for a considerable time. It is regarded as objectionable, where high pressures are employed, to restrict the opening to the gage in any manner, and it is the usual practice to install the connecting pipe without any valve or gage cock between the gage and the pump. This is done to obviate the possibility of the valve's being closed accidentally, resulting in damage to the machine.

In high-pressure hydraulic systems the pump operator, after closing the bypass valve, which causes the pressure to build up on the presses or other machines, keeps a constant watch on the gage and, when the pressure reaches the proper point, opens the bypass valve, which allows the pressure to drop without stopping the pump. In cases where the pressures are exceptionally high, as in hydraulic press work, a few extra strokes of the pump would make a considerable increase in the pressure if no outlet was provided. I have seen this method used month after month, with little if any noticeable change in the reliability of the gage. It is usually in low-pressure work that the vibrations are most noticeable, and these are generally due to defective installation or operation.

Toronto, Ont., Canada. J. T. ELTHORN.

Measurement of Air Removed from Surface Condenser

In the Nov. 18 issue A. R. Roberts inquires for a method of measuring the air discharged by a centrifugal hydraulic vacuum pump attached to a condenser. In the plant in which I am employed, we have done this very successfully.

The hydraulic vacuum pump was connected to a 22,000-sq.ft. surface condenser. A closed tank 3 ft. in diameter and 5 ft. high is used. Into the top of the tank a 4-in. pipe discharges the air from the pump through a three-way valve which directs the air into the tank or to the atmosphere by a quarter turn of the valve. A 1½-in. pipe is used to fill the tank with water, and a vent on top of the tank is used when it is being filled.

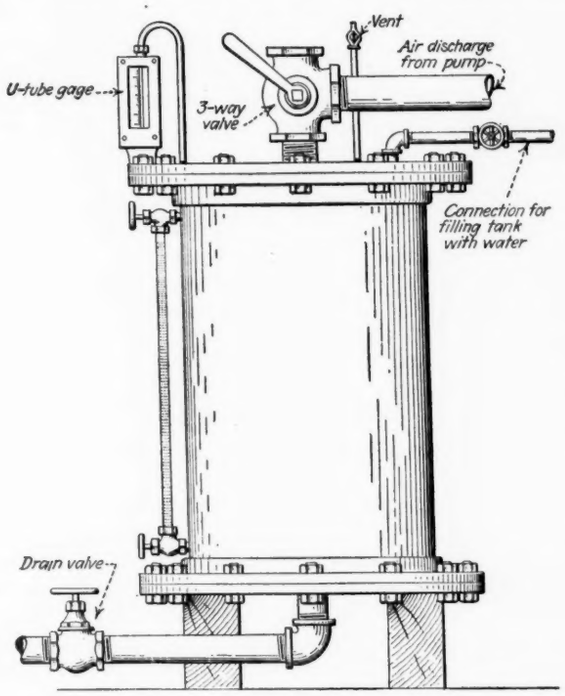
A 3-in. drain valve is connected to the bottom of the tank and a ¾-in. gage glass as near the full length of the tank as possible indicates the water level. As the desired result is the cubic feet of free air per minute, the scheme is to fill the tank with water, then turn the air from the pump into the tank and let out the water as fast as the air enters, keeping the pressure in the tank at atmospheric during this period. A "U" tube was installed on the tank so that the operator could keep the pressure constant. The tank was carefully measured to get its cubical contents and the glass graduated.

In operating the tank is first filled with water and the vent closed, then the air is turned into the tank and the drain valve opened at the same time. The man doing this, immediately adjusts the flow of water to get atmospheric pressure in the tank. This is usually accomplished before the water has fallen one foot.

As soon as the operator gives the signal that conditions are right, another man snaps a stop watch when the water reaches a predetermined mark, and again when it reaches the low mark at the bottom of the tank.

The number of cubic feet indicated on the gage glass divided by the number of seconds and multiplied by 60 would give the cubic feet per minute.

We had to make a number of changes before this outfit operated satisfactorily; the first change was to make the drain pipe larger, as the first one would not carry the water away fast enough. We also had considerable trouble with the "U" tube, which was made from a ¾-in. gage glass. We finally used a gage with a range of 6 in. which had a large chamber at one end and was filled with oil. This gave us no trouble such as was had with the glass gage using water, which frequently would be emptied when the air was first turned on. The "U" tube should be connected to the top of the tank, and the tank should never be quite full.



Arrangement of device for measuring air from condenser

This outfit is best operated by three men; one to open or close the three-way valve, another to operate the drain valve and the third to use the stop watch.

After a few trials we were able to get good results. The part requiring the most skill is the operation of keeping the drain open the right amount to keep the pressure in the tank at atmospheric, but we found no trouble in getting men to do it well.

We checked the accuracy of this method by shutting the pump off from the condenser and connecting orifices which had been calibrated to deliver a certain amount of free air.

We then measured with our outfit the air that was passed through the orifices and found that we showed an average of 6 per cent high, which in most cases is close enough when testing for leaks in condensers.

Lowell, Mass. E. T. REED.

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Cleaning Water Leg of Vertical Boiler

What method can be employed for cleaning mud from the water leg of a vertical fire-tube boiler? *J. F. W.*

After washing out all loose dirt and scale, pass a chain around the water leg and loosen the mud by drawing the chain back and forth through the water-leg cleaning handholes. A cistern chain is good for the purpose or one similarly provided at intervals with flanges that may be made from washers to act as scrapers for loosening the mud so it can be removed by flushing with water.

Steam Required to Heat Fuel Oil

In using steam at 130 lb. gage pressure to heat 16 Baumé oil from 140 to 300 deg. F., how many pounds of steam per gallon of oil are required for this purpose? *H. A. T.*

A gallon of fuel oil of gravity 16 Baumé weighs 7.992 lb., and for practical purposes is called 8 lb. The specific heat of fuel oil is 0.498 to 0.5 B.t.u. per pound, and it is usual to assume 0.5; hence to raise one gallon of fuel oil 1 deg. F. would require $8 \times 0.5 = 4$ B.t.u., or for raising its temperature through $300 - 140 = 160$ deg. would require $160 \times 4 = 640$ B.t.u. By reference to steam tables it may be found that a pound of steam at the pressure of 130 lb. gage, or $130 + 15 = 145$ lb. per sq.in. absolute, contains 1,192.8 B.t.u. above 32 deg. F. Using a pipe coil heater, the condensate discharged would contain the heat of the water, or 327.4 B.t.u. above 32; so that each pound of steam used would part with its latent heat, namely, $1{,}192.8 - 327.4 = 865.4$ B.t.u., and for heating one gallon of fuel oil from 140 to 300 deg. F. would require $640 \div 865.4 = 0.739$, or practically 0.74 lb. of steam at the pressure of 130 lb. gage. This figure, 0.739 lb. of steam per gallon of oil, does not include losses from radiation from the heater. That can be known by holding the heater filled with oil at the temperature of 300 deg. F. and weighing the amount of condensate discharged in a given time.

Air Pressure for Air Lift

What determines the air pressure that should be used for operation of an air-lift pump? *H. L.*

The air pressure at the point of delivery into the discharge pipe should be no more than necessary to insure injection of the air. Greater pressure reacts with hindrance to the flow of the ground water to the discharge pipe and the discharge, if at all, takes place in spasmodic gulps with waste of compressed air. For starting the lift, the air pressure should be practically the same as the pressure due to the head of water at rest above the lower end of the air pipe, and when in operation the air pressure should be practically the same as the water pressure due to the head of the ground water above the end of the air pipe. Hence, in starting, the air pressure should be gradually increased until a steady discharge is established, and since, in most instances, the ground-water level recedes with pumping, the air pressure should be reduced sufficiently to obtain nearly uniform discharge.

Pitting of Horizontal Return-Tubular Boiler Over Fire

We have two 16-ft. by 60-in. horizontal-return-tubular boilers that carry 90 lb. gage pressure and recently have been troubled by pitting of the boilers over the fire. We feed about 500 gal. of water per day that is subjected to a water-treatment process. How can we employ zinc plates to check the corrosion or pitting? *M. P. C.*

Zinc plates suspended in the boiler water have been used with varying degrees of success to prevent pitting by counteracting galvanic action that is productive of corrosion. The zinc usually is in the form of plates about 10x6x½ in. thick. Use about 50 such plates per boiler. The plates with separators between them may be divided into conveniently sized groups, suspended in the water of the boiler from stays. There must be good metallic contact between the zinc and steel of the boiler in order to be effective. The contact surfaces should be bright and clean and be bolted down tight to prevent scale forming at points of contact. The slabs or plates will last two or three months. It is possible that the zinc slabs will be of no benefit. Cases have been known where it was thought their presence hastened instead of retarding corrosion, therefore the progress of the corrosion should be frequently examined. It is possible that your water-treatment softening process is carried too far. A thin coating of scale or a lime wash will often act as a protection against corrosion.

Resetting Valves of Buckeye Engine

How would the valves be set on a 20x42-in. Buckeye engine? Indicator diagrams taken with average load show considerable inequality of cutoffs. *P. W.*

For correction the whole valve setting should be reviewed. Remove the valve covers so the opening and closing edges of the valves may be seen, or felt with a long piece of wire. The travel of the main rocker arm is first adjusted till it vibrates an equal distance each side of the main rocker pin. This may be checked by revolving the main eccentric on the shaft.

Adjustment to obtain this correction is made by lengthening or shortening the eccentric rod. The main valve is then squared by putting the eccentric at its greatest throw forward and noting if there is a full port opening; then putting at full throw on its opposite center and adjusting until the port will be opened a like amount on each end when the eccentric is upon that end. Any adjustment needed is made by slipping the clamp on the valve rod where the connection from the rocker arm is made to the valve stem.

The engine is then put on a dead center and the eccentric revolved in the proper direction (following the crank in this case) until the desired lead is obtained on that end. Turning on the opposite center, the lead should be the same. If not, equalize at the clamp on the valve stem. Make the lead about $\frac{1}{16}$.

To set the cutoff valve, the main eccentric and rider eccentric are placed together and the cutoff valve should then rest squarely over the ports. Placing the engine on the dead center, the governor wheel should be revolved until the throw of the eccentric is exactly in line with the crank. Secure the governor wheel in this position. Block out the governor half way and turn the engine over slowly and note when the valve is just closed on the end taking steam. Mark the distance the crosshead has traveled on the guides and turn the engine over from the opposite center and note when the valve closes on that end. If the cutoffs are not alike, find the difference between the strokes at cutoffs and put the crosshead at a point representing one-half this difference and reset cutoff valve till it just closes at that end. For example, if the cutoff from head end is found to take place at 8 inches and on the crank end at 6 inches, the difference would be 2 inches, and one-half this difference would be 1 inch. The crosshead would then be set at 7 inches, the average between the two, and the valve equalized, in that position, by lengthening or shortening the connection to the rocker arm.

Determining Safe Working Pressure for Boiler

How can I make a cold-water test of a boiler to determine the safe working pressure? *C. D.*

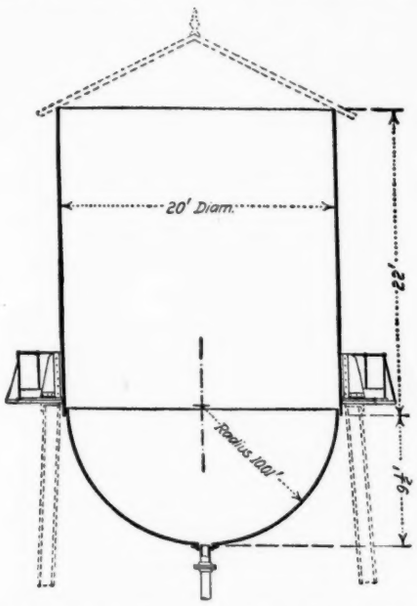
The safe working pressure is not to be determined solely by a cold water test. It must be based mainly on computation of the least safe working pressure for any part of the boiler from competent inspection of the size, arrangement and condition of tubes, stays, staybolts, rivets and shell material, and all other parts, including the character of the riveted joints of the shell, character and condition of workmanship, and also age of the boiler, and with selection of a factor of safety that conforms with good practice and is in accord with any federal, state, municipal or insurance regulations applicable to the case.

The method of making a cold-water test, called a hydrostatic test, is to fill the boiler with water to overflowing and, with the safety valve secured and all outlets closed excepting the connection with an accurate pressure gage and a hand test pump, gradually increase the pressure with the test pump, taking care that the test pressure is no more than 50 per cent greater than the safe allowable working pressure previously decided upon. The purpose of this test is not to determine the safety, but to demonstrate whether the boiler is tight for the selected working pressure or if there are any weak places that may have escaped the internal or external inspection.

Content of Tank

What quantity of water would be contained in a vertical cylindrical tank 20 ft. in diameter by 22 ft. high and a spherical bottom 96 ft. 6 in. deep, as shown in the sketch? *J. J.*

The area of a circle 20 ft. in diameter is $20 \times 20 \times 0.7854 = 314.16$ sq.ft., and the content of the cylindrical part of tank would be $314.16 \times 22 = 6{,}911.52$ cu.ft., or $6{,}911.52 \times 7.48 = 51{,}698.17$ gal. The



Principal dimensions of a cylindrical tank with spherical bottom

volume of the spherical segmental bottom would be $\left(\frac{9.5}{2} \times (20)^2 \times 0.7854\right) + \left((9.5)^3 \times 0.5236\right) = 1{,}941.2$ cu.ft., or $1{,}941.2 \times 7.48 = 14{,}520.176$ gal. Hence the total content would be $6{,}911.52 + 1{,}941.2 = 8{,}852.72$ cu.ft., or $8{,}852.72 \times 7.48 = 66{,}218.34$ gal.

Resistance of Cylindrical Shells to Collapsing Pressure

Why will not a cylindrical shell, like that of a boiler, offer as much resistance to external as to internal pressure? *J. J. B.*

A cylinder made with a shell that is thick and short in proportion to the diameter may withstand as great an internal as an external pressure. When the shell of the cylinder is thin and long in proportion to its diameter, like a boiler shell, internal pressure tends to improve the circular form; but when subjected to external pressure there is a collapsing tendency, because the initial form is not perfectly circular and the material is not absolutely uniform in thickness and strength. Consequently, theoretical formulas for the stresses in actual cylinders subjected to external pressure are of little value, since such cylinders will collapse before the compressive strength of the material is reached, and this is especially true where, like boiler shells, the cylinders are thin and long compared with the diameter.

[Correspondents sending us inquiries, should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications.—Editor.]

Operation of Three-Phase Induction Motors Under Abnormal Supply Conditions*

By D. D. Ewing and C. F. Bowman

MOTOR name-plate ratings as they apply to three-phase induction motors assume a three-phase constant voltage and constant frequency supply. In practical-motor installations these conditions are more likely to be the exception than the rule; the nominal 220-volt line may have a line-to-line voltage ranging from 180 to 250, and if the line carries a heavy, short-interval peak load, this range may be covered within the time interval of twenty-four hours, and the voltages may be badly unbalanced. The actual value of the nominal 60-cycle frequency is likely to be anything from 55 to 65 cycles; also one of the three lines to a motor may be opened accidentally, thus allowing the motor to operate single-phase.

Single-Phase Operation

Single-phase operation is ordinarily the result of the accidental opening of one of the lines to the motor. In some types of starting equipment the protecting fuses are shunted during the starting period and are in circuit only when the starter is thrown to the running position. Should one of the fuses be blown, the motor will start normally and will continue to run when the starter is thrown to the running position, but it will be operating as a single-phase motor. If the load is considerably below the rating of the motor, the motor may continue to operate in this fashion for months before the fault is discovered unless rigid inspection routines are maintained. Another common cause of single-phase operation is the burning off or loosening of the fingers or contacts on the running side of the starter. As the contacts are out of sight in the starter box or immersed in oil, this fault may also remain undiscovered for some time.

It is not always easy to detect single-phase operation except by making tests with instruments. Unusual vibration and excessive heating are two of the most common symptoms. Periodic inspection of the motors and their wiring and protective equipment is the best preventive.

Table I gives the more important descriptive data pertaining to the motors tested for single-phase operating performance. In these tests the motors were supplied from a motor-generator set which enabled the voltage and frequency to be maintained constant. Each motor was operated three-phase and the load on it adjusted until the motor drew rated current. The load was then maintained constant until thermometers, placed on the motor windings at various places to indicate temperature rise, showed constant temperature. The motor was then allowed to cool down slightly and one of the line wires was opened, causing the motor to operate as a single-phase machine. The load was then adjusted to give the same temperature rise as occurred with three-phase operation.

In Table II are shown the results of this test. It will be noticed that the horsepower output for rated current, three-phase operation, agreed closely with the horsepower ratings given by the makers in Table I. The single-phase output has a mean value, as shown by Table II, of about 50 per cent of the three-phase output, and the single-phase current per terminal for this output is approximately 20 per cent in excess of the three-phase rated current per terminal.

These data show that for lightly loaded motors single-phase operation over a long period of time is not likely to injure the motor. On the other hand, motors which are located in warm rooms and on which the mean load is in excess of 50 per cent of the rating are likely to be seriously damaged by prolonged single-phase operation.

Motor Operation at Voltages and Frequencies Other than Rated

To determine the performance of commercial motors when operated under other than rated voltage and frequency conditions, tests were run on the motors described in Table III. In making these tests the motors were all supplied from the same motor-generator set. The voltage and frequency of this set could be adjusted, and the voltage wave form was known to conform closely to that of the sine wave.

As indicated in Tables IV and V, each motor was subjected to several tests. In these tests the temperature rise was determined by means of thermometers located at various places on the windings of the motors, and the output, corresponding to the temperature rise guaranteed by the manufacturer, was determined.

It will be noticed that motors B and C were liberally rated. The tables show the effect of abnormal supply conditions on motor operation. Deviation of voltage or frequency either above or below rated values resulted in loss in horsepower capacity, the percentage decrease in capacity being roughly equal to the percentage change either way in voltage or frequency. Thus, for operation at 50 cycles, or 83 per cent rated frequency, the horsepower capacity is only about 83 per cent of the rating.

The tables do not contain data as to the effect on motor speed, and it should be emphasized that whereas voltage variations do not materially affect the speed, frequency variations do, the change being almost directly proportional to the change in frequency. Thus, a motor that runs at 1,720 r.p.m. for full load on 60 cycles, will only run at ⅚ of 1,720 or 1,430 r.p.m. or 50 cycles.

Briefly summarized the results of the tests show:

1. That three-phase motors operating single-phase may be expected to furnish safely one-half of their three-phase rated horsepower, and when so doing the current per terminal will be about 20 per cent greater than the three-phase rating.

2. Frequency and voltage variations, if not large, decrease the safe horsepower capacity by roughly the same percentage as that of their own variation from rated value.

TABLE I—DESCRIPTIONS OF MOTORS OPERATED SINGLE-PHASE

Motor Number	A	B	C	D	E	F
Rated horsepower	7.5	5.0	7.5	5.0	3.0	1.0
Rated voltage	110	110	110	110	110	110
Rated amperes per terminal	37.3	28.2	37.5	29.6	16.3	6.0
Rated speed	1700	1200*	1760	1750	1750	1700
Frequency	60	60	60	60	60	60
Guaranteed temperature rise, deg. C	.....	.....	50	50	50	50
Type of rotor	Squirrel cage	Wound rotor	Squirrel cage	Wound rotor	Squirrel cage	Squirrel cage

* Synchronous speed.

TABLE II—RESULTS FROM SINGLE-PHASE TESTS

Motor Number	A	B	C	D	E	F
Horsepower—three phase	7.45	5.08	7.23	5.12	2.89	0.95
Horsepower—single phase	4.25	2.60	4.06	2.50	1.66	0.42
Amperes per terminal, single phase	43.0	32.0	45.5	35.0	20.5	6.8
Ratio, single-phase current to three-phase current	1.15	1.14	1.21	1.18	1.26	1.13
Ratio, single-phase horsepower to three-phase horsepower	0.57	0.51	0.56	0.49	0.57	0.44

TABLE III—NAME-PLATE DATE FOR MOTORS OPERATED AT ABNORMAL VOLTAGES AND FREQUENCIES

Motor Number	A	B	C
Horsepower	5	5	7.5
Amperes per terminal	27	29.6	37.8
Voltage	110	110	110
Frequency, cycles per sec	60	60	60
Full-load speed, r.p.m	1720	1750	1705
Guaranteed temperature rise, deg. C	50	50	40
Type of rotor	Squirrel cage	Wound	Wound

TABLE IV—AMPERES PER TERMINAL TO GIVE GUARANTEED TEMPERATURE RISE FOR DIFFERENT VOLTAGES AND FREQUENCIES

Motor Number	A	B	C
110 volts, 60 cycles	27.0	35.1	45.7
110 volts, 50 cycles	26.2	37.0	49.6
110 volts, 65 cycles	27.0	37.3	44.6
90 volts, 60 cycles	27.5	35.1	45.0
120 volts, 60 cycles	26.5	35.5	43.6

TABLE V—HORSEPOWER OUTPUT FOR GUARANTEED TEMPERATURE RISE, DIFFERENT VOLTAGES AND FREQUENCIES

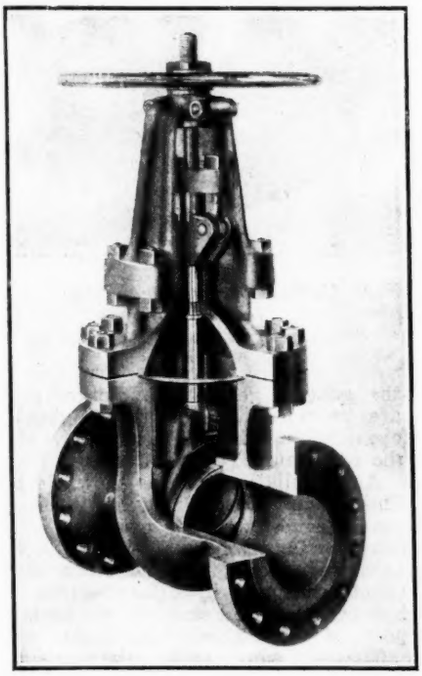
Motor Number	A	B	C
110 volts, 60 cycles	5	5.6	9.
110 volts, 50 cycles	4.25	3.9	8.5
110 volts, 65 cycles	4.9	5.6	8.4
90 volts, 60 cycles	4	4.6	7.2
120 volts, 60 cycles	5	5.5	8.0

*Abstract from a paper published in Circular No. 4, Engineering Extension Service, Purdue University, Lafayette, Ind.

New and Improved Equipment

Lunkenheimer Valves for High Pressures and Temperatures

To meet the present requirements for higher pressures, the Lunkenheimer Company, Cincinnati, Ohio, has developed a line of cast-steel monel-mounted gate, globe, check, non-return and blow-off valves for pressures of 400, 600 and 900 lb. and temperatures up to 750 deg. F. These new designs conform to the American Steel Flange Standards, formulated by the American Engineering Standards Committee.



Cast-steel monel-mounted gate valve

The illustration shows a hand-operated gate valve designed for 400 lb. 750 deg., in which several distinctive features are embodied. A similar design of motor-operated valve with remote control is included in the line. Cylindrical and spherical sections have been used wherever practical and long taping fillets provided throughout as best adapted to the higher pressures and temperatures. The bonnet bolts are double-ended studs of heat-treated alloy steel having a tensile strength of 125,000 lb. per square inch.

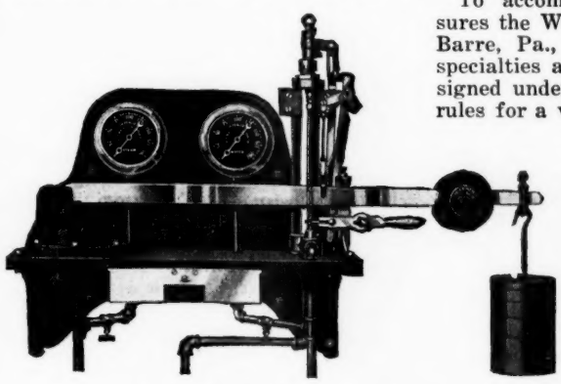
To hold the gland out of the way while repacking, two lugs have been provided on the inside of the yoke. Both the gland and stuffing box are monel lined to prevent corrosion and to permit easy withdrawal from the stuffing box when repacking. Directly beneath the stuffing box is a condensation chamber provided with a relief valve which, when opened during the repacking process and while the valves are under pressure, prevents steam escaping through the stuffing box. It also acts as a drain, should water of condensation accumulate.

The disk is of the tapered solid wedge construction and is attached to the stem by means of a U-slot on each side of the enlarged stem head. Exceptionally long machined guides are provided to insure minimum friction.

Defender Damper Regulator

The illustration shows a new-type Model A damper regulator recently added to the line of boiler-room instruments and appliances manufactured by the Defender Automatic Regulator Co., St. Louis, Mo.

The regulator consists essentially of a weighted lever that is operated by the varying steam pressure through a diaphragm, a pilot valve and operating cylinder, and an adjustable compensator.



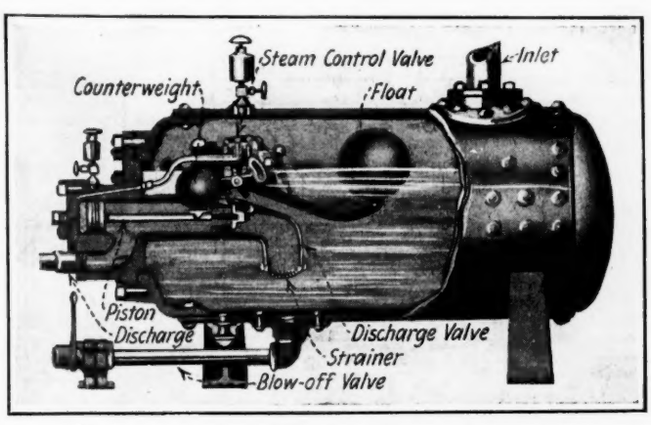
Defender model A Regulator

The weighted lever is pivoted at one end on knife-edges close to the diaphragm and connects about midway with the pilot valve, which in turn controls the water pressure admitted to the operating cylinder. The piston of the water cylinder is connected through a chain or cable directly to the damper. The compensator, which works entirely on the lever and independently of the water valve, is adjustable to work on any steam-pressure variations from 1 to 10 pounds. Provision is made for operating the pilot valve by hand and the employment of pressure gages on the regulator panel to indicate the steam and water pressure directly at the machine. The hand lever permits the damper to be opened or closed or held in any position desired, if it should be necessary to shut off the steam for repairs on the diaphragm or fulcrum bearings, etc. The water valve is constructed with a cast-brass body, having a bronze sleeve or lining that is removable and a bronze valve.

New Nicholson Steam Trap for High Pressure

The adoption of high steam pressures and temperatures, particularly in the larger central station, has made necessary important changes in boilers, prime movers and other auxiliary equipment.

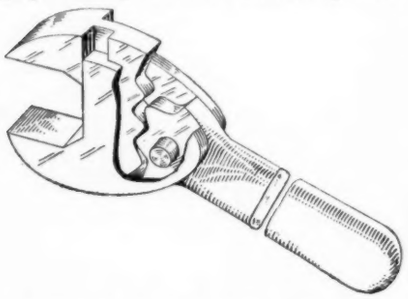
To accommodate the higher pressures the W. H. Nicholson Co., Wilkes-Barre, Pa., has added to its line of specialties a trap of the tank type designed under the A.S.M.E. Boiler Code rules for a working pressure of 600 lb. per sq.in. and a maximum temperature of 700 deg. F. The body of the trap is constructed of riveted steel plate and is fitted with inlet, outlet and blowoff connections. One end of the shell is fitted with a cast-steel cover to which is attached the operating mechanism. By removing this cover all the working parts are made accessible for examination or repairs. The principle of operation is the same as used in the regular line of traps manufactured by this company except that they have been developed to operate under higher pressures. Valves, valve seats and all wearing parts are of monel metal. The traps are built with capacities up to 17,500 lb. per hour.



Body of trap is constructed of riveted steel plate

Self-Adjusting Wrench

A self-adjusting wrench that fits any nut within its capacity is illustrated herewith. In operation the wrench is placed over the nut and a pull on the handle causes it to grip the nut, the harder the pull the more forceful the grip. The jaws are released by moving the handle in the opposite direction. It is made in three sizes: No. 1, with a range from $\frac{1}{2}$ to $\frac{5}{8}$ in.; No. 2, $\frac{3}{4}$ to $1\frac{1}{8}$ in.; No. 3, $\frac{7}{8}$ to $1\frac{3}{4}$ in.



Wrench is automatically adjustable to any size and shape of nut

The wrench is put out by the Universal Spanner Co., Dayton Bridge Road, West Ealing, London, England.

New Ash-Handling System Employs Belt Conveyor and Quenching Tanks

A new type of ash-handling system recently developed by Babcock & Wilcox, Ltd., Glasgow, Scotland, is illustrated herewith.

The complete ash-handling system is placed in the basement of the power house and consists essentially of an ash hopper and a mechanical quenching tank for each boiler, into which the ashes fall as produced. Each unit of a battery of quenching tanks delivers its quota onto one common belt conveyor in a state in which a compound rubber belt can handle them without suffering any injury and thence to a dump or overhead bunkers for further disposal.

The system is applicable for large power houses and is limited in extent only by the possible economic length of a belt conveyor, and obviously several such conveyors could be used, each with a battery of ten or more tanks.

A belt conveyor is generally accepted as an economical handling medium, and for general purposes the quenching and withdrawing methods here in use render it possible to employ such a conveyor, which hitherto has been debarred from this service.

Under each boiler a mild-steel ash hopper chute is located, this is divided into two compartments: One in connection with the front of the grate, for the fine ashes, which terminates in an open outlet and a free drop into the quenching tank; the other, for the larger ashes and clinkers from the back of the grate, has a similar delivery outlet which, however, dips for a few inches into the quenching tank and is, therefore, water-sealed. The hopper chute and the construction of the quenching tank can best be seen from the cross-sectional drawing Fig. 2.

The mechanical quenching tank is of cast iron. The side corresponding to the front of the boiler house is essentially vertical, while that corresponding to the back of the grate is at an angle of 45 deg. The base and sides of the tank are trough-shaped and form a wheel race on an arc that corresponds with the radius of the paddle wheel, which fits freely within the lower part of the trough. In this way the vanes, of which there are twelve on the wheel, rake the ashes out of the water upon the sloping side of the tank and deliver them over the side and into a shallow delivery hopper which is provided with partitions, so that the ashes are fed on to the conveyor in the direction of its running.

In order to protect the driving shaft and bearings of the paddle wheel from the injurious influences of the water, which may be contaminated and acidulous, the bearings are attached to a cast-iron bracket beyond the reach of the water in the trough. A small quantity of fresh water is constantly added, and the water level in the tank is thus maintained at a uniform height.

In order to produce the slow and intermittent rotating motion of the paddle wheel, the driving spindle is provided at its upper end with a ratchet wheel that is partly covered by a shield in which the fulcrum of the pawl is located. The shield is given a reciprocating motion by a wire rope which passes all the quenching tanks and to which each of the shields is attached.

A couple of rocking or motion posts are located one at each end of the row of quenching tanks. These are L-shaped, hinged close to the ground and connected together at their uppermost ends by the wire rope mentioned. The motion post near the driving end of the installation is coupled by a connecting rod to a crank which is part of the general gear. The motion posts are provided with counterweights in opposite directions, so that they keep the rope taut.

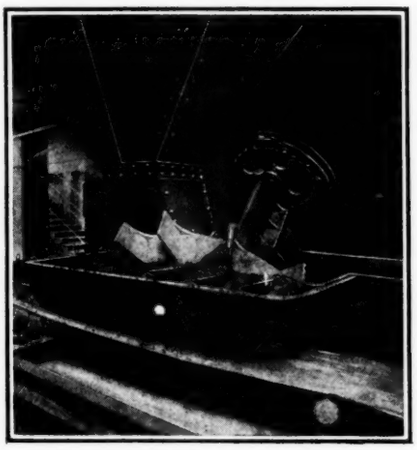


Fig. 1—One of the quenching tanks with its feeding hopper, showing delivery of ashes to the conveyor

An installation such as shown in the illustration is capable of handling one ton of ashes per hour per tank. The conveyor is driven by a 5-hp. motor, by speed-reduction gear and chain to a countershaft which, in turn, drives the belt conveyor and operates the rocking posts of the ash-quenching tanks.

The belt conveyor in this particular installation is 16 in. wide and 86 ft. long and runs at 250 ft. per min. Its upper run is supported by five-roller idlers, while the return run rests on three-roller flat idlers. The standard belt is composed of cotton rubber and has a tensile strength of 4,000 lb. per sq.in.

A conveyor of the width and speed given would be of sufficient capacity for 20 quenching tanks of the standard capacity of one ton of ashes per hour.

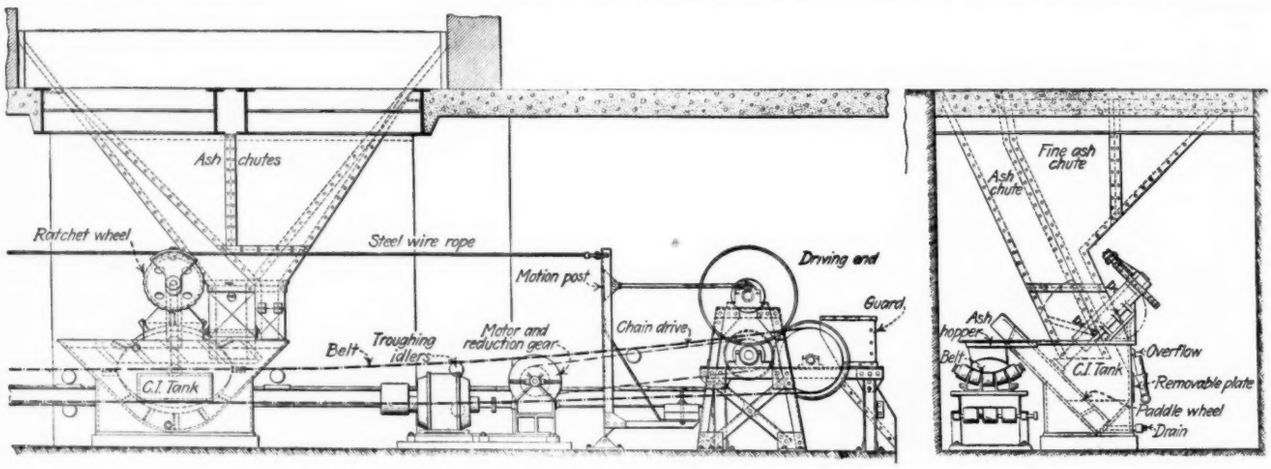


Fig. 2—Elevations of ash handling plant showing general arrangement of equipment

Operating Troubles of Brass Condenser Tubes Scientifically Investigated*

How Foreign Bodies and Air Bubbles Cause Deterioration—What Connection Chemical Composition, Low Vacuum, and Other Well-Known Factors Bear Toward Corrosion of Brass Tubes.

THE extended investigation by Guy D. Bengough and R. May is confined to the corrosion of 70:30 brass, or "Muntz metal" condenser tubes. Corrosion is studied in four principal divisions—of copper, zinc, brass, and of the condenser tubes themselves. Protection against corrosion is also taken up.

Corrosion of copper was investigated in detail so that the various chemical reactions that take place, and the arrangement of each constituent of the scale, could be fairly well defined. If commercially pure, high-conductivity copper is immersed in a dilute sodium chloride solution with free access of air, it would be expected that under highly oxidizing conditions, the scale would consist of approximately 66 per cent cupric oxychloride and 33 per cent cupric hydroxide and basic carbonate.

This, however, might be expected only at the first instant of attack in stagnant water or highly aërated high-speed streams. The first formed layer of scale would reduce the rate of supply of oxygen to the metal beneath. The result would then be cuprous oxide, Cu_2O, which is insoluble, and cupric hydroxide $Cu(OH)_2$, and basic copper carbonate, C_uCO_3, $C_u(OH)_2$. The amount of oxychloride present therefore may vary from zero up to 66 per cent, depending on the supply of oxygen.

Zinc Corrodes Less Than Copper

In practice the distribution of the corrosion products will not be uniform, and consequently the supply of oxygen which may operate will be variable. Pores in the metal which are but slightly oxygenated will be preferentially attacked. Cupric chloride, $CuCl_2$, may set up metal concentration cells, and give rise to nodules or crystals of copper among the scale constituents.

A study of the corrosion of zinc indicated that when subjected to sea water, an electrolytic zinc specimen suffered only about one-quarter of the loss that would occur in copper under similar conditions and about one-sixth of that which would be characteristic of a typical 70:30 brass tube.

Further study revealed the fact that protective scales were formed. Also, a high-speed jet of distilled water, such as 10 ft. per sec., would remove this scale to such an extent as to accelerate corrosion greatly.

The protective material contained zinc hydroxide, $Zn(OH)_2$, which is gelatinous when wet and has a remarkable power of clinging to the surface of metallic zinc from which it has been formed.

It would be expected that by using a jet of liquid, which could dissolve this scale as fast as formed, the attack would be much more rapid than under ordinary circumstances. Ammonium chloride is a liquid that will dissolve zinc hydroxide. When a water jet at 10 ft. per sec. velocity containing 5 per cent ammonium chloride, was tried it was found that intense action took place. The amount of corrosion opposite the jet was far greater than in any of the other experiments.

The corrosion of 70:30 brass was taken up from a number of angles. The metallic copper which is frequently present with dezincification proved to be that deposited from copper chlorides

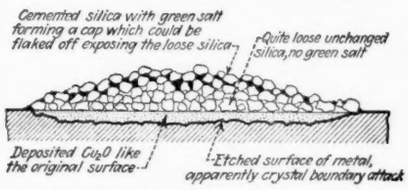

Cemented silica with green salt forming a cap which could be flaked off exposing the loose silica
Quite loose unchanged silica, no green salt
Deposited Cu_2O like the original surface
Etched surface of metal, apparently crystal boundary attack

Fig. 1—Sand restricts the oxygen supply to the tube surface, causing deposit as indicated

and not due to copper left behind when zinc is dissolved alone.

The presence or absence of copper as a corrosion product depends largely on the rate of the oxygen supply to the initial corrosion products. Where the oxygen is restricted, copper occurs. Temperature is clearly one factor affecting the presence of copper, since the solubility of oxygen falls off as temperature rises.

Zinc chloride, $ZnCl_2$, is a clear liquid, which often oozes out of minute black specks or pores in the metal. This is protective, keeping the surface of the brass bright and yellow. The reason has not here been investigated. Scale formed by the crystallization of the liquid thickens and tends toward being opaque with deposits of copper very finely crystalline spread beneath it. In this way a plug or layer of copper is begun.

If foreign matter is present, the supply of oxygen may be restricted so that metallic copper may appear in the form of a loose layer, precipitated from its chloride. If, on the other hand, the scale consists entirely or mainly of the zinc oxysalt which formed a blister at the mouth of the pore from which the zinc chloride escapes, the action is more localized and the copper tends to be precipitated close to the brass surface. This forms a porous plug beneath which the action is carried out locally and plug dezincing occurs. Copper plugs contain some cuprous oxide, $C_{u_2}O$, indicating slow penetration of oxygen.

Arsenic is found to be active in preventing corrosion. The precise way in which it accomplishes this is not yet known.

Corrosion taking place under foreign material, such as loose sand, is illustrated in Fig. 1. It is believed that the true explanation of corrosion taking place beneath foreign material is that the supply of oxygen is restricted. It appears that oxygen must penetrate slowly, since the metal formed is cuprous oxide, $C_{u_2}O$, and not a cupric compound. The cuprous oxide, together with cupric chloride, C_uCl_2, is no doubt formed from the decomposition of cuprous chloride in conditions of low oxygen concentration.

Some of the cuprous chloride diffuses outward and eventually reaches positions where increased concentration of oxygen exists. There it changes to cupric oxychloride and cupric chloride.

The former is precipitated among the granules of the deposit, so that often the outer layers of the sand, or fine material, are cemented together. Cuprous oxide is deposited before leaving the surface of the metal, as indicated in the figure.

Tube Corrosion in Laboratory

Condenser-tube corrosion represents the largest section of this report, covering deterioration from physical defects, chemical composition, water, air bubbles and other factors. In some parts of the investigation experiments were carried on in a model condenser with sea water, etc.

It was found possible to produce in the laboratory most of the phenomena observed in corroded tubes. It has been established that the circular cones or plugs formed in dezincing are not due to unmixed copper, but rather to that which has been redeposited from copper chloride solution.

Tubes that do not show copper plugs at any stage of their career, are ordinarily those containing more than about 0.01 per cent arsenic, the exact amount depending partly on the presence of other metals. Those that do show it, consist of arsenic-free brasses; highly purified brass readily shows it, but particularly brasses containing iron and manganese.

Dezincification is decreased by arsenic, tin, nickel, aluminum, tungsten and lead, in this order of relative importance. It is increased by iron and manganese.

Arsenic is by far the most powerful of all metals and can easily neutralize the effect of ten times the amount of iron. Other factors that influence de-

*Abstract of Seventh Report of the Corrosion Research Committee of the Institute of Metals, by Guy D. Bengough and R. May, delivered at the September 8-11 meeting in London.

zincification are temperature conditions, water speed, scales or membranes at the metal surface, and the presence of pores in the metal.

Increase of temperature up to at least 50 deg. C., or 122 deg. F., increases the rate at which dezincing develops. The speed of the cooling water affects not only the temperature, but also the rate of oxidation; a high water-speed tends to restrain dezincing.

A thin layer of any material that tends to keep dissolved copper in close contact with the metal surface and oxygen away from it tends to encourage the formation of copper deposits. Cheese cloth, for instance, will encourage dezincing. Slimy deposits, such as mud, iron compounds, animal and vegetable organisms, also encourage it.

It is often found that brass contains minute pores. These represent the start of the formation of a plug of copper. Pores in arsenic-bearing tubes appear to set up pinhole pits containing cuprous oxide.

Arsenic Free Tubes Desirable

The scale of arsenic-free tubes is ordinarily very tenacious, being white or gray in color. It is hard to knock off the surface of the metal by a stream of aërated sea water, at least as high as 10 ft. per second. Tubes carrying this type of scale have been known to have a useful life of more than twenty years in marine condensers and of ten years in estuarine power stations.

There seems to be a distinct future for arsenic-free tubes in modern plants, with ample steam inlet space, high vacuum and high water-speeds. A minimum arsenic-bearing tube of less than 0.01 per cent has never yet shown copper deposition even in stagnant water, in the authors' experience. In modern condensers with high vacuum and water speeds, an arsenic-free tube,

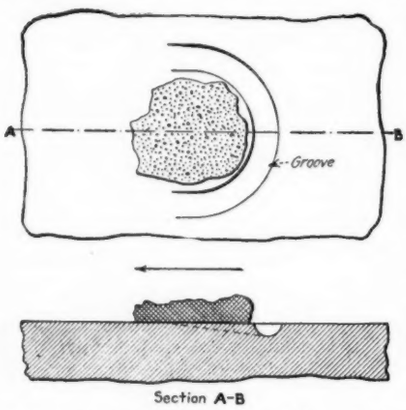


Fig. 2—Foreign body on tube surface causes groove corrosion

low in iron, should form a scale high in zinc compound of highly protective value.

It would not be advisable to specify an arsenic-free tube and then place it in a condenser using a low vacuum and low water speed coupled to a reciprocating engine. Corrosion is here restricted because of higher temperatures.

With respect to physical conditions it has been shown that the surface layer of worked brass known to be free from oil and similar in composition to the underlying material is the more resistant to sea water corrosion. A result of inspections of tube corrosion indicates that surface imperfections do not initiate rapid corrosion.

After an extended investigation involving large numbers of tubes, marking of flaws, and recording their effects with regard to corrosion, the authors conclude that only rarely do such flaws produce corrosion or pitting. The majority of corrosion troubles are due to causes that can be demonstrated at will in a laboratory on selected portions of tubes quite free from visible and probably invisible flaws. Flawed tubes are likely to suffer undue losses by separating into layers, when corrosion has got well under way. They should be rejected on these grounds.

The effect of crystal size as investigated agrees in general with the results of Shaw Sparrow. When tubes are of strictly comparable compositions, fine-grained tubes are preferred to coarse. In ordinary practice fine-grained tubes are of different composition.

Low Vacuum and Corrosion

The fact that pitting of condenser tubes seems to be greatly increased since the advent of the steam turbine appears to be due to the lower temperatures as compared with those previously utilized in the 26-in. vacuum, or less, of the steam engine. In the case of the turbine, comparatively low temperatures, around 95 deg. F., are encountered, while in the steam engines temperatures of 130 to 140.

It appears that the higher temperatures are in the region of the death point of abundant forms of bacterial life. The production of ammonia, oxygen and in some cases sulphuretted hydrogen is attributed to these organisms. At low temperatures, therefore, their products are more active in forming corrosion. Oxygen, also, is less soluble at the higher temperatures.

Pitting of condenser tubes, which sometimes takes place near the inlet and sometimes at a distance from the inlet such as six inches or two feet, has been well demonstrated as being due to conditions of the flow of water and presence of air bubbles.

Water entering a tube in a state of disturbance gradually changes, giving place to a normal type of flow which is turbulent above the critical velocity or smooth below that velocity. Sea water circulated through the tubes of an experimental condenser at from five to eight feet per second, entered in an abnormally turbulent state, but attained the "Reynolds," or normal, state of turbulence after going two or three feet from the inlet.

During the first part of the flow there is an abnormal impingement of rapidly moving water and air bubbles. Small bubbles move as part of the water without abnormal contact, but large bubbles tend to travel together. They enter together and are rapidly broken up, interfering with the nearly stationary viscous film and the water in the tube.

Helical motion of the water is often caused by ferrules at the ends of the tubes. This induces air bubbles first to collect in the center of the entering water stream and then to be distributed against the tube, where the more normal flow later develops. Thus corrosion at a point from six inches to three feet from the inlet of the tube is readily explained owing to the spin that the water receives on entering. It has been found that filling the notches of

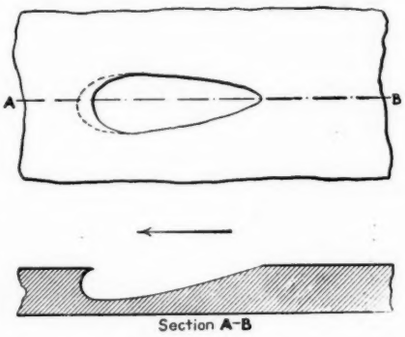


Fig. 3—Bright pits from air bubbles with tails upstream

the ferrule up with wax eliminated the helical motion in some cases of water entering tubes.

A foreign body attached to the wall of the tube will cause erosion in the normal type flow, as indicated in Fig. 2. A viscous film traveling next to the tube goes more slowly than the main stream of water. This strikes the obstruction and tends to flow around it, causing a curved groove, as indicated. After the groove has been formed, the original obstruction may be removed and corrosion will continue.

It has been found that sea water with entangled air is active in this form of corrosion. Water free from air may allow a protective scale to form. Air bubbles may start corrosion, after which the effect of the viscous film may continue it, with results frequently noted in condenser tubes.

Where Air Troubles Attack

The effect of air bubbles would be to attack the first pass of the condenser near the inlet. The second pass of a condenser tube would be more inclined to show at some distance from the inlet or near the outlet. The reason is that in the first case, air bubbles are irregular, and in the second they are more uniformly diffused through the water, becoming effective only after considerable travel. Mechanical obstructions on the tube surface are largely responsible for air-bubble corrosion in the second pass.

Corrosion from air results in bright pits as in Fig. 3. A tail forms on the upstream side and a sharp edge downstream. These have not been found produceable in stagnant water. With arsenic-free tubes, violent air-bubble action is required, or else a protective scale will form, preventing such pitting.

Types of scales encountered are: (1) The calcium carbonate type; (2) the vitreous protective type; (3) the unoxidized copper type, usually white or gray, with copper crystals in lower layers; (4) the crystalline partly oxi-

dized copper type, usually blue, with a red layer below; and (5) the crystalline fully oxidized copper type, usually blue and green with little or no red layer.

The rate of penetration of oxygen through these scales is least in the case of No. 1 and increased up to No. 5. In case No. 1 it may be so slow that practically no corrosion takes place. With scale No. 5 oxygen penetration may be so rapid and uniform that in some cases thinning of the tube takes place.

Scales of type No. 1 are commonly met with in hard fresh water, but are rarely associated with corrosion. No. 2 is the most dependable of all scales and is usually thinner than No. 1 and may exercise great protective powers. Scales of the type of No. 3 are almost as protective as those of No. 1. The scales in connection with tube composition are studied at great length.

It is clear that protective scales on arsenic-free Admiralty and lead-brass tubes may be preferred. It appears that the presence of cuprous oxide in the lower layers of a scale is usually a sign that heavy corrosion of the tube will develop sooner or later. An investigation of the relation of electrical factors to corrosion indicates that there is no definite connection.

The subject of tube protection by the formation of suitable scales is being studied. Tube cleaning may be a vital factor in prolonging the life of the tube where corrosive scales are formed, or it may be objectionable if carried out too thoroughly in the presence of protective scales.

Protective scales, it is believed, can be formed in the presence of sea water. Those produced by certain types of dock water have proved to be very resistant when subjected to air jets and water. Imitations of the scale formed by sea water to be used in fresh-water conditions have been attempted but not yet been satisfactorily concluded.

Runner and Guide Vanes Damaged on Large Waterwheel

RECENTLY, the runner and guide vanes of one of the 70,000-hp. turbines in plant No. 3B of the Niagara Falls Power Co. were damaged by a piece of timber or other obstruction lodging in the runway, to such an extent that it was decided advisable to replace the injured parts with spares, which were available. Although the turbine that was damaged had been in operation for ten months without trouble, it is probable that when the third 70,000-hp. unit was placed in operation, the increased velocities carried objects to the wheels which previously had remained undisturbed. In this plant it was found advisable to operate without racks in the penstock intakes on account of their depth below the surface of the forebay.

Subsequent to the accident, timbers of the character of railroad ties and logs as much as eleven inches in diameter had been found in the wheel casing. It is believed that a log of this type entered through the guide vanes, was caught by the runner buckets and dragged around the periphery of the runner, breaking the shearing links on most of the guide vanes almost simultaneously. It is of course impossible to say how much of the breakage was due directly to the obstruction and how much was a secondary action caused by broken pieces of guide vanes or by the runner vanes striking the guide vanes after these were pushed against the runner or left free to strike the runner. It is a condition of units having very large water passages that an object may pass into the turbine, having a resisting strength which increases disproportionately to the strength of the turbine parts with which it may come in contact.

The shearing links are provided on most large turbines and are designed to fail at a predetermined opening or closing pressure if a foreign object becomes lodged between the guide vanes. This saves the remainder of the operating mechanism from undue strain and results in the failure being on the outside of the turbine where it can be cheaply and easily repaired. It is quite a common occurrence to have links break in this way in waterwheel operation. When a single link breaks, the guide vane will seek a position not far from the closed position. Furthermore, the two adjacent guide vanes will prevent the free vanes from coming in contact with the runner should it for any reason tend to swing to a position where it might be hit by the runner. This however, is not the case when several consecutive links are broken, since the vanes are then unrestrained and some of them may come in contact with the runner. The breakage of more than one link at a time is something that never occurred before in the operation of the company's plants.

The company had on hand a spare runner, which by co-operation in design between the Wm. Cramp & Sons Ship & Engine Building Co. and the Allis-Chalmers Manufacturing Co. can be used in any one of the three 70,000-hp. machines. Two of the three wheels were built by the former company and one complete unit and the spare runner by the latter.

After the accident an inspection showed that the damage had been confined to the entrance edges of the runner vanes and the edges of the guide vanes, no injury having been sustained by the casing, distributor plates, draft tube, runner crown or band, or by the guide-vane stems, and no effect being produced on the runner clearances or on the bearings for the guide vanes or the mainshaft bearing. No cracks were found in the fillets joining the runner vanes to the crown or band, and except for the pieces broken out of the entrance tips of the vanes, the runner casting was found to be in good condition, and it has already been repaired by welding in new pieces at the tips. At the time of the accident the unit continued to carry about 45,000 kw. until it was shut down by closure of the Johnson valve, this closure being completed within four minutes from the time the first rumbling was heard, indicating the presence of the obstruction in the turbine. Owing to the manner in which the operators handled the situation, there was no interruption of service and the switches on the unit were not pulled until the Johnson valve had reduced the load to a low value. The localization of damage in the turbine guide vanes and runner speaks well for the mechanical construction of the unit.

As a result of study of the problem encountered in this installation, it has been decided to equip all the turbines with mechanical stops to reduce as far as possible the action of guide vanes swinging into contact with runner vanes after an initial breakage due to an obstruction. It is believed that these means will have a tendency to localize the breakage and confine it to the replaceable links.

Credit is hereby acknowledged to the late John L. Harper, who was chief engineer and vice-president of the Niagara Falls Power Co., and H. Birchard Taylor, vice-president of the Wm. Cramp & Sons Ship & Engine Building Co. for the information in this article.

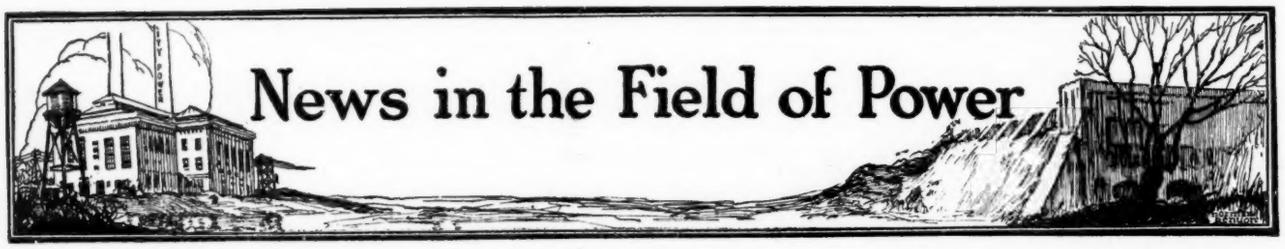
Efficient Utilization of North Dakota Lignite

That the lignite of North Dakota may be burned much more economically than the present practice, generally speaking, is the conclusion of George B. Wharen, in presenting a report on "Efficient Methods for Burning North Dakota Lignite for Steam Generation," which appeared in *The Quarterly Journal* of the University of North Dakota, Vol. 14, No. 4.

A study of the most economical equipment for consuming the North Dakota lignite, which averages approximately 33 per cent moisture and 6,800 B.t.u. per lb., reveals that the application of some better engineering practice and methods of operation would do much to improve many of the present plants.

Rocking grates afford better opportunity to work the fire than do stationary grates, and their use lengthens the period between cleanings, but produces slightly more combustible loss in the ash. Best efficiency is at present obtained from traveling grates with non-sifting stokers, and ignition arches, for burning small sizes of raw lignite with probably a combined boiler and furnace efficiency of 68 to 70 per cent by the year. Skilled operators are required for efficient handling representing an undesirable condition.

A study of the application of pulverizing indicates that with the unit system and waste heat drier there is every indication that the cost of producing steam would be 30 per cent less than with the ordinary hand-fired furnace and improved none-sifting grates. Operating costs should be lower than with sprinkler-type stokers and would at least equal that of the traveling grate. Conditions generally are briefly and satisfactorily outlined, giving much information obtained directly from operating installations.



Seabury Elected Secretary of the A. S. C. E.

George T. Seabury, manager of the Providence Safety Council and former division engineer of the Providence Board of Water Supply, has been elected secretary of the American Society of Civil Engineers. Mr. Seabury was named to fill the position left vacant through the death in July of John H. Dunlop, at the meeting of the Board of Direction of the society at its recent meeting at Pittsburgh.

Failure of Casting Costs Engineer His Life

Pickens O. Ladd, assistant engineer at the Colorado State Museum, was instantly killed on the morning of Nov. 28, by steam, due to a failure in the steam line when he opened the valve of the steam-heating plant. An investigation by the boiler inspector revealed that among the causes of the accident were water hammer causing the failure of a defective casting and the insufficient anchoring of the steam line.

Engineering Sessions Planned for A.A.A.S. Meeting

The seventy-ninth annual meeting of the American Association for the Advancement of Science will be held in Washington Dec. 29-Jan. 3. Two sessions will be devoted to engineering, one on Tuesday and the other on Wednesday evening, Dec. 30 and 31. The first meeting will be devoted to the general topic of engineering research and the second meeting to some topic of general interest to engineers such as the relation of engineering to the fundamental sciences.

Power Plant Equipment Exports Show Increase

As compared with a year ago the exports from the United States during October of direct-current generators of under 500-kw. capacity were practically doubled, while shipments of large direct-current generators increased over 50 per cent, according to government reports. The same percentage increases apply to alternators.

Exports of steam-turbine generator sets in October were in value less than half the total for September, though the number was more than double, indicating that the bulk of this business was in very small units.

Following the gain in generating equipment, exports of transformers increased approximately 40 per cent, as compared with the preceding month, the figures being $346,368 for October and $248,011 for September.

Long Beach Plant Built in Record Time

After establishing a speed record for construction, the Southern California Edison Co. placed the first 35,000-kw. unit of its Long Beach steam plant on the line Dec. 3. Originally planned for completion in fourteen months, work on the plant was rushed on an emergency schedule due to the power shortage brought about by a drought, with the result that the first unit was brought into service eleven months after excavation was started.

Contributing to this feat was a record-breaking transcontinental shipment consisting of twenty-two freight cars which made the trip from Schenectady to Long Beach in eight days. Every modern device for efficiency and economy in steam generation has been used in the new plant with the result that it will produce 350 kw.-hr. per barrel of oil, in comparison with 225 kw.-hr. per barrel in the original Long Beach plant.

With the placing in service of the second unit in the plant within the next few weeks and the installation of an additional 55,000 hp. to the Big Creek hydro-electric development, now under construction, the generating capacity of the Edison system will be brought up to a total of 672,000 hp.

State Anthracite Tax Again Upheld by Supreme Court

The United States Supreme Court without handing down a written opinion affirmed the judgment of the Supreme Court of Pennsylvania on the constitutionality of Pennsylvania's anthracite tax. This decision was the outcome of an appeal of the Cranberry Coal Co. and others from a decision of the lower courts in favor of the state.

At its last term the Supreme Court upheld the constitutionality of this tax on anthracite levied by the State of Pennsylvania, which had been challenged by a number of mining companies on the ground that it was a tax on interstate commerce. In that opinion the court held that coal is a local commodity until placed on cars for interstate shipment, and that the tax was constitutional because it applied on coal before being loaded.

Following this decision, a number of anthracite operators brought a new test case, the one just presented the Supreme Court. In effect the basis of this action was that the state tax was applied by the weight of the coal; that it was not weighed as raised at the mines, because no scales for the purpose being available, it was weighed in the cars after having been loaded for interstate shipment, therefore giving it an interstate-commerce nature. The lower courts held that the time of weighing was a matter of convenience and an incident.

World Petroleum Production 1,018,900,000 Barrels

World petroleum production passed the billion barrel mark for the first time last year, it was shown recently in government figures, and production in the United States attained a new record.

The total world output reached 1,018,900,000 bbl. of 42 gal. each, according to statistics compiled by the Geological Survey. The United States produced 732,407,000 bbl., which was 71.9 per cent of the total world output. Mexico, second largest producer, had an output of 149,585,000 bbl., or 14.7 per cent of the world total.

World production in the 66 years since 1857 has aggregated 11,388,626,000 bbl. of which the United States has produced 63.2 per cent.

Dr. Durand Presented With New Gavel

The Student Branch of the A.S.M.E. of the Pratt Institute of Brooklyn presented, with appropriate ceremonies, to Dr. William F. Durand, the new president of the A.S.M.E., a gavel, made at the Institute, on which a silver band had been placed with Dr. Durand's name, the name of the society and his date of office engraved upon it. Dr. Durand was inducted into office on Tuesday, Dec. 2, at the annual meeting of the A.S.M.E., this meeting having the largest registration, 2,174, of any so far held by the society.

Doctor Durand, who was professor emeritus of Leland Stanford University, has had a long service as a teacher of engineering. He taught mechanical engineering at the Michigan State Agricultural College, then went to Cornell, where he was professor of marine engineering until 1904, and from there in the same year to Leland Stanford, where he was head of the mechanical engineering department. He is a graduate of the United States Naval Academy and during the war rendered noteworthy service on the National Advisory Committee for Aëronautics and on the National Research Council, as well as doing effective work later as a member of the Interallied Commission on War Inventions abroad.

Doctor Durand is a member of the A.S.N.A. & M.E., A.I.E.E. and other scientific societies. He is a recognized authority on marine engineering and has written extensively on this as well as other engineering subjects.

The society is to be congratulated on having such a distinguished engineer to lead it for the coming year.

Winnipeg Power Plant Plans Completed

Plans for a $3,000,000 power plant at Slave Falls on the Winnipeg River and a $2,000,000 transmission line to Winnipeg have been completed by the Winnipeg Hydro-electric System according to press reports. Work will begin as soon as the Norman Dam control problem is satisfactorily settled.

Senate Will Probably Pass New Muscle Shoals Bill

As this is written, it seems apparent that the Senate will pass, before this will appear in print, the Underwood Muscle Shoals bill as modified to meet suggestions of the Secretary of War. The Underwood measure is regarded primarily as a government-ownership proposition. It does provide for the lease of the properties to private enterprise, but the conditions are such as to make such disposition unlikely. It is very apparent that the Senate does not expect a lessee to come forward, as the greatest care is being exercised in perfecting the government - ownership alternative.

Strangely enough, the House, which voted overwhelmingly for the Ford bill, is expected to vote down the bill which the Senate is expected to pass. Many members of the House voted for the Ford bill because they were anxious that no further government money be expended at Muscle Shoals. The Senate is strengthening the mandatory operation feature of the Underwood bill and with each amendment is making it less acceptable to the other chamber. The probabilities all are that the House, the membership of which is overwhelmingly against public excursions into commercial business, will vote down the measure.

John E. Bell Dies on November 27

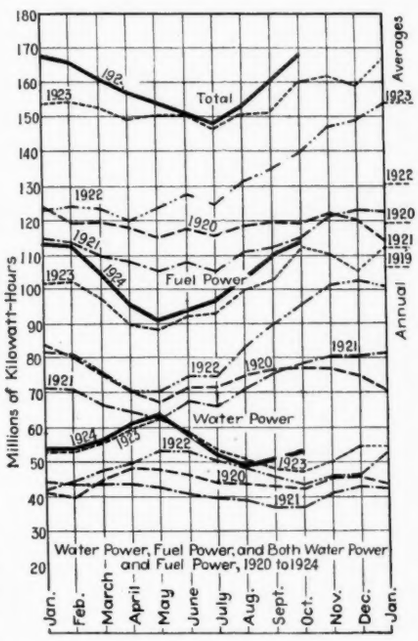
Thanksgiving morning John Everett Bell, consulting engineer specializing in the power-plant field, died suddenly of pneumonia and the industry lost one of its efficient and practical engineers. Born in Ripley, Ohio, in 1876, he studied in the common schools and after one year at the Ohio State University entered the drawing room of L. L. Summers in Chicago. From there he went to work for the Florence & Cripple Creek R.R. and then to the R. E. LaBelle Mill & Water Power Co. where he became superintendent, later becoming connected with the Stirling Boiler Co., of Chicago, where he had charge of a great deal of the technical and engineering work. During this time he was particularly interested in the thermodynamic end of engineering, and was in a measure responsible for the bringing out of the Stirling book on steam, which, although in the form of a catalog, was in reality a textbook and the first book of this nature to be issued by a boiler company.

After 1907 the Babcock & Wilcox Co. consolidated with the Stirling Co. and Mr. Bell moved to the New York office of the Babcock & Wilcox Co. During the time he worked with them, he made a large number of tests involving the study of transfer of heat for boilers, superheat and furnace work.

About ten years ago he left this company to establish a consulting practice of his own, in which he was very successful, as his steam studies and experience made his services of value to the firms he represented. Some of these firms were: The Edge Moor Co., where he worked on waste heat boilers; the Power Specialty Co., on superheater and other boiler work; the Combustion Engineering Co., on furnace design for boilers; the Sinclair Oil Co. During this work Mr. Bell developed a special type of waste-heat boiler suitable for cement-kiln practice. He is survived by his widow.

Production of Electricity Increases

The average daily production of electricity by public-utility power plants in October was 167,300,000 kw.-hr., 4½ per cent greater than the daily output for the month of September and practically equal to the record rate of production established in January. The curve of total daily output for 1924 indicates that the increase in the use of electricity since July is continuing at a nearly uniform rate. If this rate of increase is continued to the end of the year, the output of electricity by public-utility power plants will start off in the new year at about normal.



Power curve goes up

The average daily production of electricity by public-utility power plants and the proportion produced by water power for each month of this year were as follows: January, 167,400,000 kw.-hr.—32.3 per cent; February, 166,700,000 kw.-hr.—32.3 per cent; March, 160,800,000 kw.-hr.—34.4 per cent; April, 158,000,000 kw.-hr. — 39.2 per cent; May, 154,600,000 kw.-hr.—40.7 per cent; June, 151,800,000 kw.-hr.—37.5 per cent; July, 148,800,000 kw.-hr.—34.8 per cent; August, 152,700,000 kw.-hr.—32.0 per cent; September, 160,100,000 kw.-hr.—31.1 per cent; October, 167,300,000 kw.-hr.—31.4 per cent.

Diversion Case Before the Supreme Court

Arguments in the appeal of the Chicago Sanitary District from a decision of the Federal District Court granting the United States an injunction against the diversion of water amounting to 10,000 cu.ft. per sec. from Lake Michigan when only 4,167 cu.ft. were authorized in the permit issued by the Secretary of War under authority of Congress, were presented before the United States Supreme Court Dec. 8 and 9.

Members of the court evidenced keen interest in the issues presented, and counsel were interrupted by frequent questions from the bench. From the tenor of these question the indications are that the Supreme Court will sustain the injunction, about the only point that appears in doubt being the time that will be granted the Chicago district to make other arrangements for disposal of its sewage before the injunction shall be fully operative.

Inasmuch as the Supreme Court has a voluminous record and many briefs to examine, an early decision is not expected. Meanwhile, an effort is being made to have Congress enact legislation to authorize Chicago to divert 10,000 sec.ft. and the subject is being negotiated between Canada and the United States in connection with the St. Lawrence survey.

Bureau of Standards Needs New Power Plant

The annual report of the Bureau of Standards, signed by George K. Burgess, contains a recommendation for a new central power plant to supply the Bureau. It says:

"The Bureau of Standards is the place above all others where one would naturally look for the most efficient plant for supplying heat, light and power. Instead, the bureau has been forced for several years to get along with a very inefficient and decentralized mechanical plant, consisting of the original power plant designed to supply two buildings, two additional boiler plants for heating, and numerous pieces of electrical machinery scattered through the various buildings for converting energy brought from the power company's plant to energy suitable for use in the laboratories.

"The efficiency of such an installation is necessarily very low; more coal is consumed and more men are required to operate and care for the machinery than would be the case in a modern central power plant.

"In addition, there are certain disadvantages brought about through the peculiar nature of the Bureau's work. For supplying the needs of a scientific laboratory power supply must above all things be continuous. This is never the case when power is bought from a commercial concern and transmitted over a long distance. Often valuable results are lost through failure of the power, and the fact that this has to be taken into consideration in designing a large part of our laboratory apparatus greatly complicates the work."

American Petroleum Institute Discusses Oil Supply

At the American Petroleum Institute's fifth annual meeting, which was held in Fort Worth, Texas, Dec. 9-11, the statement was made, according to the press, by the president of the Institute, T. A. O'Donnell, in his address that properly developed, the oil supply of the country will last for 100 years. Other estimates, it is reported, notably that of H. L. Doherty, place the date of exhaustion at 12 years hence. He said that he favors a law that would place oil prospecting on the same basis as irrigation and reclamation projects are now handled by districts, counties or communities which would have authority to hold drilling of wells within reasonable bounds.

Engines and Waterwheels Manufactured in 1923

The Department of Commerce announces that, according to the data collected at the biennial census of manufacturers, 1923, the establishments engaged primarily in the manufacture of engines and water wheels reported products valued at $266,997,778, an increase of 33.8 per cent as compared with 1921, the last preceding census year. This industry classification covers all establishments whose principal products were steam engines (other than locomotives) and turbines, internal-combustion engines of all kinds, tractors and traction engines, and water wheels, motors, and turbines, produced for sale as such. The figures here given do not, therefore, include data in regard to engines made in motor-vehicle factories for installation in complete automobiles, trucks, motorcycles, etc., manufactured in the same factories.

In 1921 the industry was represented by 296 establishments, the decrease to 249 in 1923 being the net result of the loss of 72 establishments which had been included for 1921 and the addition of 25 which had not been included for that year. Of the 72 establishments lost to the industry, 24 had gone out of business, prior to the beginning of 1923, 18 were idle during the entire year, 26 had been engaged primarily in the manufacture of engines or water wheels in 1921 but reported other commodities.

Commerce Department Budget Decreased for Coming Year

The budget carries $22,741,514 for the work of the Department of Commerce during the next fiscal year. This is $1,200,000 less than the amount actually appropriated for the department for the present fiscal year. The total is made up as follows: Office of secretary, $230,380; contingent expenses, $200,000; rent, $68,000; printing and binding, $500,000; Bureau of Foreign and Domestic Commerce, $2,914,864; Bureau of Census, $1,974,000; Steamboat Inspection Service, $1,057,470; Bureau of Navigation, $526,240; Bureau of Standards $1,750,410; Bureau of Lighthouses, $9,677,980; Coast and Geodetic Survey, $2,298,230; Bureau of Fisheries, $1,543,940.

The budget for the Bureau of Standards carries certain appropriations for research and special work of interest to *Power* readers as follows: For apparatus, $70,000; for fuel, heat, light and power, $43,500; for the maintenance and operation of testing machines, $38,000; for the investigation of public-utilities standards, $100,000; for the technical processes used in the manufacture of clay products, $27,000; for high-temperature investigations, $9,740; for the standardization of electrical and mechanical devices and other equipment, $110,000.

The number of acres of public land reserved as valuable for water-power purposes and held subject to disposal under the Federal Water Power Commission totaled 4,762,844 acres at the end of the last fiscal year, according to information made public at the Interior Department. These lands are along rivers and streams in twenty states and territories.

DETAILED STATISTICS OF ENGINES AND WATERWHEELS 1923 AND 1921

Class and census year	Total number	Number, by indicated porsepower capacity: Under 50	50 to 100	100 to 500	500 to 1,000	1,000 to 5,000	5,000 and over	Total indicated horsepower rating	Value
1923a	739,308	729,030	7,006	2,792	256	157	67	16,024,921	$266,997,778
1921a	372,415	366,605	3,367	2,038	179	170	56	8,143,886	$199,565,239
Steam engines (not including traction engines):									
Stationary, reciprocating—									
1923	4,039	2,255	1,532	363	70	14	..	266,790	$7,187,065
1921	2,023	1,401	159	399	53	11	..	180,908	$6,202,974
Stationary turbines—									
1923	1,709	901	283	429	30	21	45	1,507,456	$7,084,676
1921	1,421	636	243	412	40	61	29	926,694	$8,711,769
Marine, reciprocating—									
1923	589	475	100	5	5	4	..	35,185	$779,440
1921	1,558	1,390	100	11	11	40	6	192,545	$4,836,944
Marine turbines—									
1923	.....	.....	....	....	....	..	..	.....	.....
1921	18	8	....	....	....	1	9	269,990	$2,139,213
Diesel—									
1923	1,313	448	351	425	69	20	..	199,161	$13,250,513
1921	785	546	60	125	39	15	..	96,116	$8,345,701
Industrial combustion engines—Stationary—									
1923	118,282	116,403	857	1,013	9	..	..	694,172	$22,409,667
1921	103,782	103,032	348	402	..	..	..	524,144	$18,185,966
Water turbines and motors—									
1923c	527	212	48	117	30	98	22	709,495	$5,604,014
1921	2,940	2,676	48	134	30	40	12	473,421	$3,218,606

a The statistics for 1923 do not include data for establishments with products valued at less than $5,000. The figures for 1921, however, include data for 27 such establishments, with an aggregate output valued at $66,664.

c Not including water motors driven by water from street mains.

Chicago A.S.M.E. To Hold Power Meeting

On January 14 and 15, 1925, the Chicago Section of the American Society of Mechanical Engineers is to hold another Power Meeting. The first meeting, held a year ago, was such a decided success from the viewpoint of attendance and interest in the proceedings, that the Power Meeting has become an annual affair.

Two days will be devoted to the coming meeting. An excellent selection of papers will be presented on subjects of current interest in the power-plant field. An entire session on prime movers will be of particular interest. Other papers on air preheating, cooling furnace walls, combustion control and auxiliaries give such a variety of interest that invitations will be extended to surrounding sections, so that the meeting will take on the aspects of a regional affair.

The features will be a banquet with music and special entertainment and an inspection trip through the new Crawford Avenue station of the Commonwealth Edison Co. With its high-pressure boilers, steam reheating and its large turbines of three different types, the plant should prove a decided attraction. An entire afternoon will be devoted to its inspection.

A tentative program of the papers to be presented follows: "Combustion Control," by T. A. Peebles, Hagan Corp.; "Large Steam Turbines," by Francis Hodgkinson, Westinghouse Electric & Manufacturing Co.; "Uniflow Versus Compound Duoflow Engines," by Robert Cramer, consulting engineer, of Milwaukee; "The Diesel Engine," by L. H. Morrison, *Power;* "Preheating Furnace Air," by C. W. E. Clark, Dwight P. Robinson & Co.; "Boiler and Turbine Room Auxiliaries" (author to be announced later); "Features of the Crawford Avenue Station," by W. L. Abbott, Commonwealth Edison Co.

Personal Mention

Sir Dugald Clerk was recently presented a Royal Medal, by the president and council of the Royal Society, for his application of scientific principles to engineering problems, particularly to the development of the internal-combustion engine.

Lawrence W. Wallace, executive secretary of the American Engineering Council, is making a month's tour of the South and West, during which he will address local engineering societies interpreting the aims of the Council and stressing the movement to establish a National Department of Public Works.

A. F. Greaves-Walker, of Raleigh, N. C., has been appointed consulting engineer for the United States Bureau of Mines. He will co-operate with engineers from the Columbus section of the bureau now engaged in investigating the refractories requirements of large steam power stations. Mr Greaves-Walker recently became head of the ceramics department of the North Carolina State College.

Society Affairs

The Birmington Section of the A.S.M.E. will hear J. Frank Rushton speak on "The Effect of Refrigeration Upon Civilization," at its Dec. 16 meeting.

The Metropolitan Section of the A.S.M.E. will at its Dec. 16 meeting hear George Van Tuyl, of the Power Specialty Co., speak on "Reducing Steam Costs by Improving Furnace Design."

The Boston Society of Civil Engineers will hear A. C. Eaton, hydraulic engineer, Power Construction Co., Worcester, Mass., give an illustrated talk on "The New England Power Co. Davis Bridge Development," at its Dec. 17 meeting.

The Combined Associations of Greater New York of the N.A.S.E. will hold a meeting on Dec. 17, at the Engineering Societies Building, 29 West 39th St., New York City. Addresses will be delivered by Fred R. Low and Frederick Felderman, and Charles M. Ripley will give a lecture illustrated with lantern slides.

Business Notes

The Link-Belt Co., 910 South Michigan Ave., Chicago, Ill., has recently completed, for its Detroit office a new building and warehouse at 5938 Linsdale Avenue.

W. Horace Williams Co., Inc. is the name of a new organization of engineers and general contractors who will specialize in design of dams, buildings, industrial plants, etc. The office will be located at 816 Howard Ave., New Orleans, La.

The Sanford Riley Stoker Co., Worcester, Mass., is enlarging and improving its Detroit plant by an addition to the main erecting shop, to meet the demand for combustion apparatus for the larger-sized boilers, also an addition to the foundry.

The Fulton Iron Works Co., St. Louis, Mo., announces the appointment of A. G. Wright, formerly with McIntosh & Seymour, to take charge of its branch office in Dallas, Texas. The Fulton-Diesel sales territory handled from Dallas includes the State of Oklahoma as well as Texas.

The Jacobson Engineering Co., consulting engineers, Oak Grove at West 15th St., Minneapolis, Minn., specializing in the design and supervision of construction of hydro-electric power developments, pulp and paper mills, announces the removal of its offices to 461 Northwestern National Life Building, Minneapolis, Minn.

The Perolin Co. of America, 1112 West 37th St., Chicago, Ill., boiler metal treatment, announces the appointment of F. D. Harger, for several years connected with the Mono Corp. of America and later with the C. J. Tagliabue Manufacturing Co., as manager and supervising engineer of its Eastern division with offices at 342 Madison Ave., New York City.

Coming Conventions

American Association for the Advancement of Science. Burton E. Livingston, Smithsonian Institution Washington, D. C. Meeting at Washington, Dec. 29-Jan. 3.

The American Boiler Manufacturers Association. H. N. Covell, 191 Dikeman St., Brooklyn, N. Y. Midwinter meeting at Cleveland on Feb. 12.

American Ceramic Society. R. C. Purdy, Lord Hall, Ohio State University, Columbus, Ohio. Convention at Columbus, Ohio, Feb. 16-21.

The American Engineering Council. L. W. Wallace, 24 Jackson Place, Washington, D. C. Annual meeting at Washington, D. C., Jan. 16-17.

American Institute of Electrical Engineers. F. L. Hutchinson, 29 West 39th St., New York City. Midwinter convention at New York City, Feb. 9-13.

American Institute of Mining and Metallurgical Engineers. F. F. Sharpless, 29 West 39th St., New York City. Annual meeting at New York City, Feb. 16-19.

American Society for Testing Materials. C. L. Warwick, 1315 Spruce St., Philadelphia, Pa. Annual meeting at Atlantic City, N. J., June 22-26.

American Society of Heating and Ventilating Engineers. F. C. Houghten, 33 West 29th St., New York City. Annual meeting at Boston, Jan. 28-30.

Engineering Institute of Canada. Fraser S. Keith, 176 Mansfield St., Montreal, Que. Annual meeting at Montreal, Que., Jan. 27-29.

Exposition of Inventions—American Institute, E. W. Bartlett, 47 West 34th St., New York City. Exposition at Engineering Societies Bldg., 29 West 39th New York City, April 27 to May 2, formerly announced for Dec. 8-13.

Illinois Society of Engineers. E. R. Tratman, Wheaton, Ill. Meeting at Chicago, Jan. 14-16.

Iowa Engineering Society. J. S. Dodds, Box 202, Ames, Iowa. Convention at Des Moines, Iowa, Jan. 27-30.

Master Boiler Makers Association. H. D. Vought, 26 Cortlandt St., New York City. Convention at Chicago, May 19-22.

The National Association of Stationary Engineers. F. W. Raven, 417 South Dearborn St., Chicago, Ill. National convention and exhibition at St. Paul, Minn., Aug. 31-Sept. 4. Annual conventions and exhibitions of state associations are scheduled as follows: **Pennsylvania Association** at Pittsburgh, Pa., May 15-16. Frank J. McCarron, 3647 North 11th St., Philadelphia, Pa. **New Jersey Association** at Atlantic City, June 4-8. H. W. Vail, 1244 Park Ave., Plainfield, N. J. **New York Association** at New York City, June 11-13, W. T. Meinzer, 3rd St., near Warburton, Bayside, L. I. **New England States Association** at Worcester, Mass., June 18-20. F. L. Tyler, 32 Briggs St., Taunton, Mass. **Connecticut Association** at New Haven, June 25-27. George F. Klopfer, 30 East Pearl St., New Haven.

National Electric Light Association. M. H. Aylesworth, 29 West 39th St., New York City. Annual convention at San Francisco, June 15-20.

National Marine Engineers Beneficial Association, Geo. A. Grubb, 313 Machinists Building, Washington, D. C. Convention at Hotel Franklin Sq., Washington, D. C., Jan. 19.

Society of Automotive Engineers. C. F. Clarkson, 29 West 39th St., New York City. Annual meeting at Detroit, Mich., Jan. 20-23.

Society of Industrial Engineers. George C. Dent, 608 South Dearborn St., Chicago, National convention at Hotel Winton, Cleveland, Ohio, May 6-8.

Western Association of Electrical Inspectors. G. M. Miller, 1304 South 7th St., Louisville, Ky. Annual convention at the Brown Hotel, Louisville, Ky., Jan. 27-28.

Trade Catalogs

Valves—Sarco Co., Inc. "Sarco Packless Inlet Valves" is the title of a leaflet recently issued by the company.

Surface Air Cooler—General Electric Co., Schenectady, N. Y. Bulletin No. 45609 describes the principles and construction of this surface air cooler as well as the arrangement of control instruments.

Grates—Neemes Foundry, Inc., Troy, N. Y. Bulletin No. 101, "Superior Dumping Grate," describes the hand dumping grate built by this company. Bulletin No. 102, "Improved Shaking and Dumping Grate," describes another type of hand-operated grate manufactured by them. Both bulletins are well illustrated with photos of the entire grate as well as the parts.

Thermometer — American Schaeffer & Budenberg Corp., Berry and South Fifth Sts., Brooklyn, N. Y. "Crescent and other Thermometers for all Industrial Requirements," is the title of Catalog No. 200, issued by the company, which gives a comprehensive description of these instruments and their uses. It is well illustrated and contains tables of dimensions and price lists.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal, net tons.	Market Quoting	Dec. 1, 1924	Dec. 8, 1924
Pool 1	New York	$3.00	$3.00
Smokeless	Columbus	2.25	2.25
Clearfield	Boston	2.25	2.35
Somerset	Boston	2.35	2.40
Kanawha	Columbus	1.65	1.65
Hocking	Columbus	1.75	1.75
Pittsburgh No. 8	Cleveland	1.90	1.90
Franklin, Ill.	Chicago	2.50	2.50
Central, Ill.	Chicago	2.25	2.25
Ind. 4th Vein	Chicago	2.50	2.50
West Ky.	Louisville	1.75	1.75
S. E. Ky.	Louisville	1.75	1.75
Big Seam	Birmingham	1.90	1.90

FUEL OIL

New York—Dec. 11, light oil, tank-car lots; 28@34 deg. Baumé, 5c. per gal.; 36@40 deg., 5¼c. per gal., f.o.b. Bayonne, N. J.

St. Louis—Nov. 25, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.85 per bbl.; 26@28 deg., $1.90 per bbl.; 28@30 deg., $1.95 per bbl.; 30@32 deg., $2 per bbl.; 32@36 deg., gas oil, 5c. per gal.; 38@40 deg., 6c. per gal.

Pittsburgh—Nov. 26, f.o.b. local refinery; 30@34 deg., fuel oil, 5c. per gal.; 36@40 deg., fuel oil, 5¼c. per gal.

Dallas—Dec. 6, f.o.b. local refinery, 26@30 deg., $1.20 per bbl.

Philadelphia—Nov. 25, 28@30 deg., $2.10@$2.163 per bbl.; 18@20 deg., $1.87½@$1.93¾; 13@16 deg., $1.764@$1.827 per bbl.

Boston—Dec. 8, tank-car lots, f.o.b.; heavy oil, 12@14 deg. Baumé, .044c. per gal.; light oil, 28@32 deg. Baumé, 5¼c. per gal.

Cincinnati—Dec. 9, tank-car lots, f.o.b. local refinery, 24@26 deg. Baumé, 5¾c. per gal.; 26@30 deg., 5¾c. per gal.; 30@32 deg., 6¼c. per gal.

New Plant Construction

Ark., Batesville—The city is having plans prepared by V. V. Long & Co., 1300 Colcord Bldg., Oklahoma City, Okla., and will soon receive bids for extensions to the water-works system. Project will include 1,000,000 gal. per min. hydraulically operated filter plant, motors, engine, etc.

Calif., Blythe—The city plans to construct a sewage pumping station, consisting of reinforced concrete pump, two 4 in. electric sewage pumps, switches, etc.

Calif., Long Beach—Pacific Electric Ry. Co., 6th and Main Sts., has awarded the contract for the construction of a 1 story, 33 x 43 ft. electric sub station, to Witt & Chute, 2516 West Bardana St., Los Angeles. Estimated cost $18,000.

Calif., Los Angeles—Los Angeles Investment Co., 1016 South Bway., is having plans prepared by Walker & Eisen, Archts., 701 Great Republic Life Bldg., for a 12 story, 85 x 135 ft. store and office building at 10th St. and Bway. Estimated cost $600,000.

Calif., San Francisco—Rousseau & Rousseau, 1171 Market St., plan to construct a 16 story Class A hotel at Market and 8th Sts.; estimated cost $1,500,000.

Conn., Hartford—Moses Fox Co., 543 Prospect Ave., has awarded the contract for a store and office building at Pratt and Trumbull Sts., to James C. Stewart, Inc., 17 East 42nd St., New York; estimated cost $800,000.

Fla., Deland—The city voted $800,000 bonds for municipal improvements, part of which will be used for the construction of a new lighting plant.

Ill., Chicago—Hooper & Janusch, Archts., 879 North State St., are receiving bids on separate contracts for a 14 story apartment hotel for the Wellington Arms Co., A. Wills, Pres., 7631 East Lake Terrace; estimated cost $3,200,000.

Ill., Chicago—Northwestern University, c/o W. D. Scott, Pres., Evanston, is having plans prepared by J. Gamble Rogers, Arch., 367 Lexington Ave., New York City, and Lowe & Bollenbacher, Assoc. Archts., 108 South La Salle St., for a group of college buildings on East Chicago Ave. near Michigan Blvd.; estimated cost $10,000,000.

Ill., Chicago—20 East Elm Street Corporation, 20 East Elm St., is having plans prepared by W. P. Doerr, Archt., 28 East Jackson Blvd., for a 17 story apartment hotel; estimated cost $1,000,000.

Ill., Galesburg—The Rowe Manufacturing Co. is in the market for 2 horizontal pumps, 18 to 24 in., for pumping water out of river, total lift 24 ft. Oil engines to work with the pumps have already been purchased.

Ia., Glenwood—The City, B. N. Maxwell, Clk., plans to hold an election Dec. 29 to vote $20,000 bonds for improvements to the water-works. Project will include either a filtration plant of 300,000 gal. daily capacity or deep well supply, pump house and tiling.

La., Eunice—The Town plans to construct a waterworks system and a light plant; estimated cost $25,000. S. Wyble is mayor.

La., Glenmore—The Town, A. G. Pringle, Mayor, plans to expend $12,500 for the installation of a water-works system and light plant.

La., New Orleans—Board of Port Commissioners, J. H. Walsh, Genl. Mgr., Suite 200, New Courthouse Bldg., will receive bids until Dec. 18 for three 500 kva. power transformers.

Me., Portland—Desmond & Lord, Archts., 1 Beacon St., Boston, Mass., and J. H. De Sibour, Assoc. Archt., Edmonds Bldg., Washington, D. C., are preparing plans for an 11 story hotel at Congress and High Sts. for a corporation composed of Washington, New York and Boston capitalists. Estimated cost $2,500,000.

Mass., Boston—The Keystone Manufacturing Co., 53 Wareham St., is in the market for motor generator sets from 10 to 30 kw., 220 volts, single and 3 phase.

Mass., Boston—Hotels Statler, E. M. Statler, Pres., Hotel Pennsylvania, 7th Ave. and 33d St., New York, will soon award the contract for a 12 story hotel; estimated cost, $4,000,000. G. B. Post & Sons, 101 Park Ave., New York, are architects.

Mass., Cambridge—S. Lilly, 21 Elmwood Park., Malden, and J. Silverman, 55 Hamilton St., Haverhill, plan to construct an apartment building at Massachusetts Ave. and Chauncey St.; estimated cost $500,000.

Mass., Lynn—The Lynn Gas & Electric Co. plans extensive improvements to its power plants on Broad St., West Lynn. Estimated cost $2,000,000.

Mo., Appleton City—The City voted $53,000 bonds for water-works improvements, including 2 deep wells, pumping equipment, etc. Burns & McDonnell, 402 Interstate Bldg., Kansas City, are engineers.

Every one of these items is reported by our authorized correspondents who are instructed to verify every item sent in. This free weekly service is published in the interests of the buyer and the seller, to bring them together and get machinery moving.

Everything possible is done to insure authenticity and timeliness.

Your co-operation is invited in helping us maintain this service at the highest efficiency.

BUSINESS NEWS DEPARTMENT
Tenth Ave. at 36th St., New York

Mo., Boonville—The Missouri Light & Power Co., A. W. Mattison, Genl. Mgr., has purchased a 20 acre site on the river east of Boonville and will construct a light and power plant to have a unit capacity of 1,000 kw. Estimated cost $200,000.

Mo., Fredericktown—The City voted $112,000 bonds to be used for new water-works and distribution system. C. Haskins, 517 Finance Bldg., Kansas City, is engineer.

N. J., Atlantic City—W. G. Souders, 31 Nassau St., New York, plans to construct an 11 story hotel at Albany Ave. and the Boardwalk. Estimated cost $5,000,000.

N. Y., New York—R. Darrow, c/o H. J. Krapp, Archt., 114 East 16th St., is having plans prepared for a 15 story hotel at 224 West 49th St.; estimated cost $1,200,000.

N. Y., New York—Fred T. Ley Co., 19 West 44th St., plans to construct a 20 story bank and office building at 54th St. and Madison Ave. Will award separate contracts.

N. Y., New York—T. C. Pennly Co., Inc., 370 7th Ave., has awarded the contract for an 18 story office and warehouse building at 9th Ave. and 34th St., to Starrett Bros., 101 Park Ave.; estimated cost $4,000,000.

O., Massillon—The Central Steel Co., R. E. Bebb, Pres., has awarded the contract for the steel work for the proposed blast furnace and power plant, to Riter-Conley Co., Oliver Bldg., Pittsburgh, Penn. Total estimated cost of entire project about $10,000,000.

Okla., Chickasha—The Chickasha Ice & Cold Storage Co., plans to enlarge its plant and install new equipment.

Okla., Pawhuska—W. D. Beard, Pawhuska, plans to construct a 1 and 2 story ice plant; estimated cost $75,000. Work will be done by day labor.

Okla., Perry—The City is in the market for a 500 hp. Diesel oil engine and electric generator for the city light plant.

Okla., Wetumka—The City will receive bids after Dec. 30 for improvements to the water-works system, estimated cost $52,000; also for improvements to electric light plant, estimated cost $60,000. V. V. Long & Co., Colcord Bldg., Oklahoma City, are engineers.

Tex., Ft. Worth—Ft. Worth Club Building Assn., A. G. Carter, Chn. Bldg. Com., has awarded the contract for the construction of a 12 story, 100 x 200 ft. club and office building on 7th St., to Bellows & McClay, First Natl. Bank. Estimated cost $1,250,000. Contract awarded on guaranteed cost, plus fixed fee to contractors.

Tex., Galveston—G. W. Frazer, 2327 Bway., will build a new ice factory and 2 story cold storage plant; estimated cost $42,000.

Tex., San Antonio—Greyburg Oil Co., F. L. Thompson, Pres., is having preliminary plans prepared for a 10 story office building at Houston St. and Broadway; estimated cost $700,000.

Wis., Cumberland—The Fame Canning Co., 111 West Washington St., Chicago, Ill., awarded the contract for the construction of a cannery, including a power house, to Nyman Construction Co., Cumberland; estimated cost $125,000. Canning and conveying machinery will be required.

Ont., East York—The Township is having plans prepared by G. R. Jack, Engr., Municipal Yard, Sammon Ave., Toronto, for sewage disposal plant, including air compressor, motor, pump, valves, etc. Estimated cost $100,000.

Ont., Ft. William—The City plans to extend its sewerage system and install sewage pumping plant, and is interested in prices of materials and equipment. Estimated cost $45,000. C. B. Symes, City Hall is engineer.

Ont., Sarnia—Hydro Electric Power Commission of Ontario, 190 University Ave., Toronto, has purchased site for transformer and distribution station and will have plans prepared for same. Estimated cost between $50,000 and $100,000. F. A. Gaby is chief engineer. Transformers, switches, etc., will be required.

Ont., Sault Ste. Marie—The City plans to purchase and install gasoline driven auxiliary pumping unit, about 1000 Imp. gal. per min., 3 stage, centrifugal, also 150 hp. gasoline engine.

Que., Quebec City—LeTramway de Quebec., c/o Rene Dupont, Quebec City, will soon receive bids for a new electrical development.